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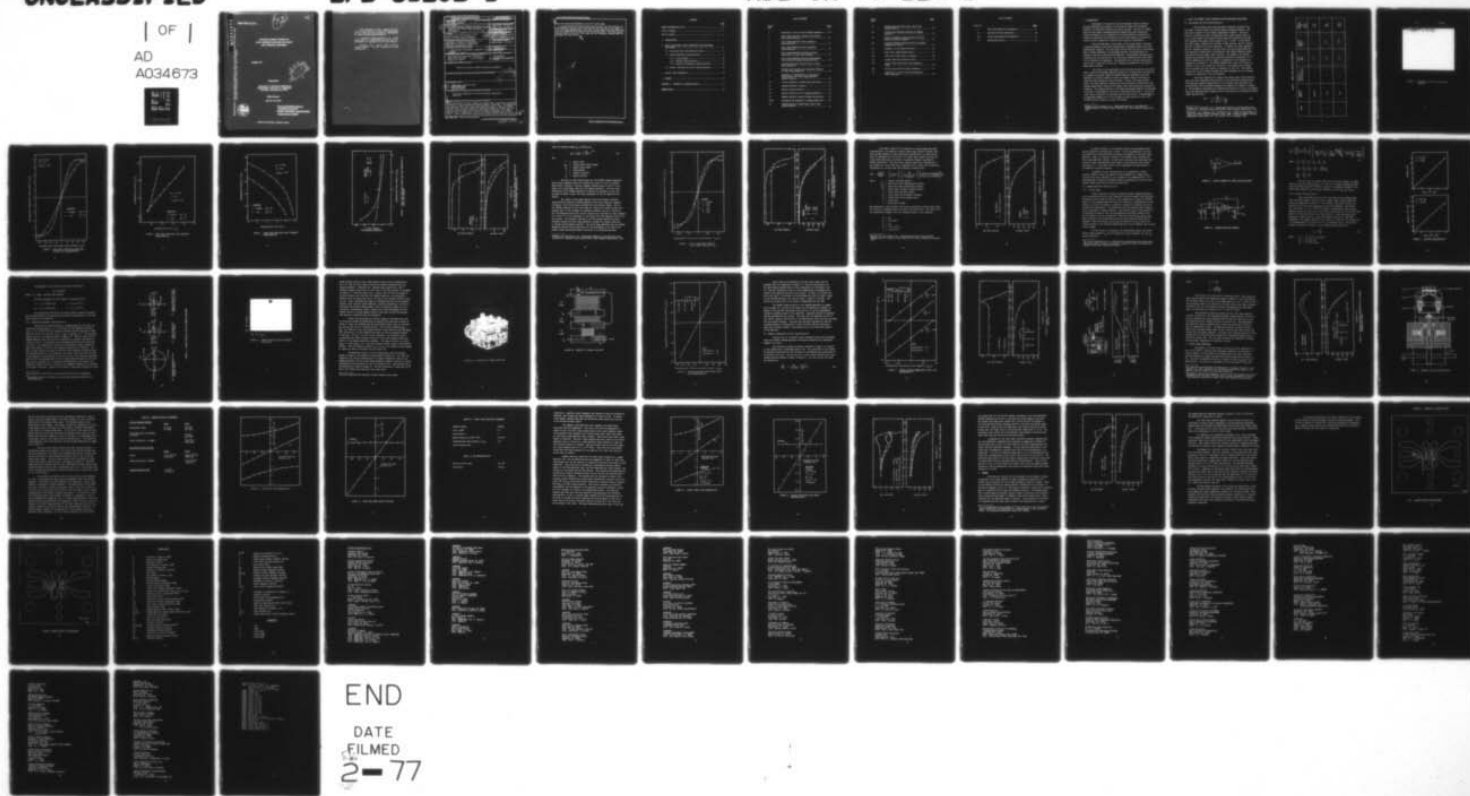
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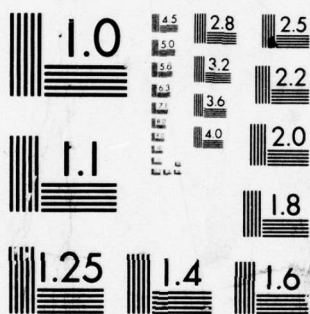
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HDL-CR-76-223-1, Multistage Hydraulic Summing and Signal Processing Amplifiers and Fluidic Input Servovalve Development, by David Lee, David M. Wornaley



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**MULTISTAGE HYDRAULIC SUMMING AND
SIGNAL PROCESSING AMPLIFIERS AND FLUIDIC
INPUT SERVOVALVE DEVELOPMENT**

OCTOBER 1976

Prepared by

Department of Mechanical Engineering
Massachusetts Institute of Technology
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U.S. Army Material Development
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20. ABSTRACT (Continue on reverse side if necessary and identify by block number) The development of fluidic summing and dynamic signal compensation operational amplifiers for high performance hydraulic systems is described. The amplifiers are constructed from three stage fluid amplifiers using as input and feedback elements linear resistors constructed from small diameter to length ratio tubing and capacitors constructed from flexible metal diaphragms. The summing amplifier has a pressure gain of 50 and a frequency response which is flat to 100 Hz. Dynamic compensation elements were constructed with lead-lag action.		

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where the lead action dominates in the 20 to 40 Hz range.

A four-way fluidic input servovalve has been developed. The valve consists of a fluid amplifier driving a pair of bellows that act directly on the flapper of the first stage flapper-nozzle valve, and a second stage spool valve whose position is controlled by the output of the first stage valve. The valve accepts pressure inputs in the 0 to 7 kPa range and has a frequency response which has decreased by 3 dB at 30 Hz.

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1. INTRODUCTION

Development of effective pure fluid sensing, signal processing and power amplification elements can effectively extend the performance capability of hydraulic actuation and control systems. The use of a common fluid for both signal sensing/processing elements and power modulation/actuation elements and the elimination of electro-hydraulic interfaces increases the potential for application of pure fluid systems in systems subjected to severe temperature, radiation and vibration environments.

The work described in this report is the second phase of research to develop hydraulic pure fluid sensing-signal processing elements and power amplification elements. The first phase of this work¹ has led to the development of design techniques for multistage hydraulic fluid amplifiers for pressure, flow and/or power gain applications. These design techniques were illustrated in the construction of a three stage amplifier with output power of 42 W from HDL 2-2B laminates. The amplifier has a blocked load pressure gain of 325, a power gain of 825 and a frequency response with less than 90° phase shift at 80 Hz.

The focus of the second phase of work has been directed to two areas. First, using the multistage amplifier design as a starting point, multiple input summing and dynamic signal processing hydraulic amplifiers have been developed to provide a basis for static and dynamic signal processing required in high performance control systems. Second, a four-way, fluidic input, two stage hydraulic servovalve has been developed to provide the link between signal power levels and the higher power level modulation required for actuators. The characteristics of the signal processing elements are described in Section 2; development of the valve is described in Section 3. In all the tests results described, the working fluid has been Univis J-43 maintained at an operating temperature of 27°C at which the fluid density is $8.69 \times 10^2 \text{ N-s}^2/\text{m}^4$ and the viscosity is $1.88 \times 10^{-2} \text{ N-s/m}^2$.

¹Wormley, D.N. and Wilson, D.R., "Multistage Hydraulic Fluid Amplifier Characteristics," Technical Report HDL-TR-156-1, Harry Diamond Laboratories, 1974.

2. STATIC AND DYNAMIC SIGNAL PROCESSING USING MULTISTAGE GAIN BLOCKS

2.1 Multistage Gain Block Characteristics

The multistage gain block described by Wormley and Wilson¹ has been used to develop static and dynamic signal processing elements. The gain block consists of three stages, summarized in Table I. The first stage is constructed of HDL 2-2B laminates with a supply nozzle width of $b_s = 0.5$ mm, whereas the second and third stages are constructed with HDL 2-2B laminates with supply nozzle widths of $b_s = 1.0$ mm.¹ In the second and third stages, laminates were stacked to form sections, and the sections were put in parallel to form an amplifier. To improve the input impedance and the frequency response of the gain block, the first stage design has been modified by replacing the 2-2B laminates with 231004A laminates, while the second and third stages remain the same. The design of the 231004A laminate was performed using the techniques described by Drzewiecki et al.² The two laminates are shown in Figure 1; dimensions of the laminates are summarized in Appendix A. The 231004A laminate has control ports that are reduced in size at the supply port interface, a new supply nozzle design and an increased size vent region in comparison to the 2-2B. This smaller control port in the 231004A increases its input impedance and its frequency response in comparison to the 2-2B laminate.

Test data comparing the blocked load pressure gain characteristics, input and output impedance characteristics, bias sensitivity characteristics and frequency response characteristics of the single stage amplifiers constructed from 2-2B and 231004A laminates are summarized in Figures 2 through 6. The parameters specifying the configurations tested are cited in the figures where the operating supply pressure level is specified by the modified Reynolds number N'_R defined as:

$$N'_R = \frac{N_R}{\left(1 + \frac{1}{\sigma}\right)^2 \left(1 + \frac{x_{th}}{b_s}\right)} \quad (1)$$

¹ Wormley, D.N. and Wilson, D.R., "Multistage Hydraulic Fluid Amplifier Characteristics," Technical Report HDL-TR-156-1, Harry Diamond Laboratories, 1974.

² Drzewiecki, T.M., Wormley, D.N., and Manion, F.M., "Computer-Aided Design Procedure for Laminar Fluidic Systems," Trans. ASME, Journal of Dynamic Systems, Measurement, and Control, Vol. 97, Series G, No. 4, December 1975.

TABLE I. THREE STAGE AMPLIFIER PARAMETERS

Stage	Laminate Design	Number of Sections to Form Amplifier	Size, b_g (mm)	Section Aspect Ratio, σ	Supply Pressure, P_s (kPa) [psi]
1	231004A	1	0.5	2	110 [16]
2	2-2B	2	1.0	0.5	661 [96]
3	2-2B	8	1.0	0.25	3970 [576]

2-2B

231004A

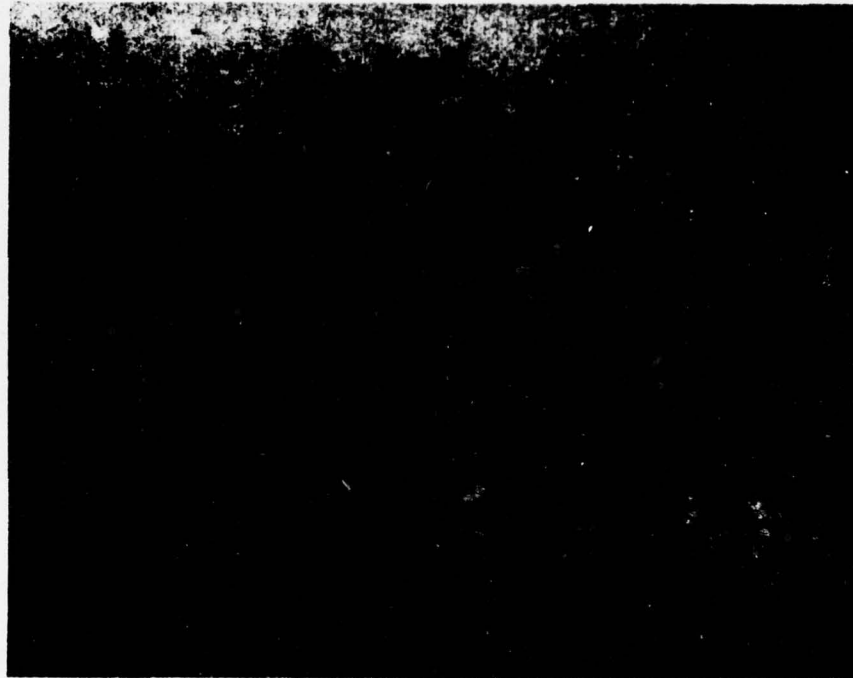


FIGURE 1. PHOTOGRAPH OF THE 2-2B AND 231004A
LAMINATES

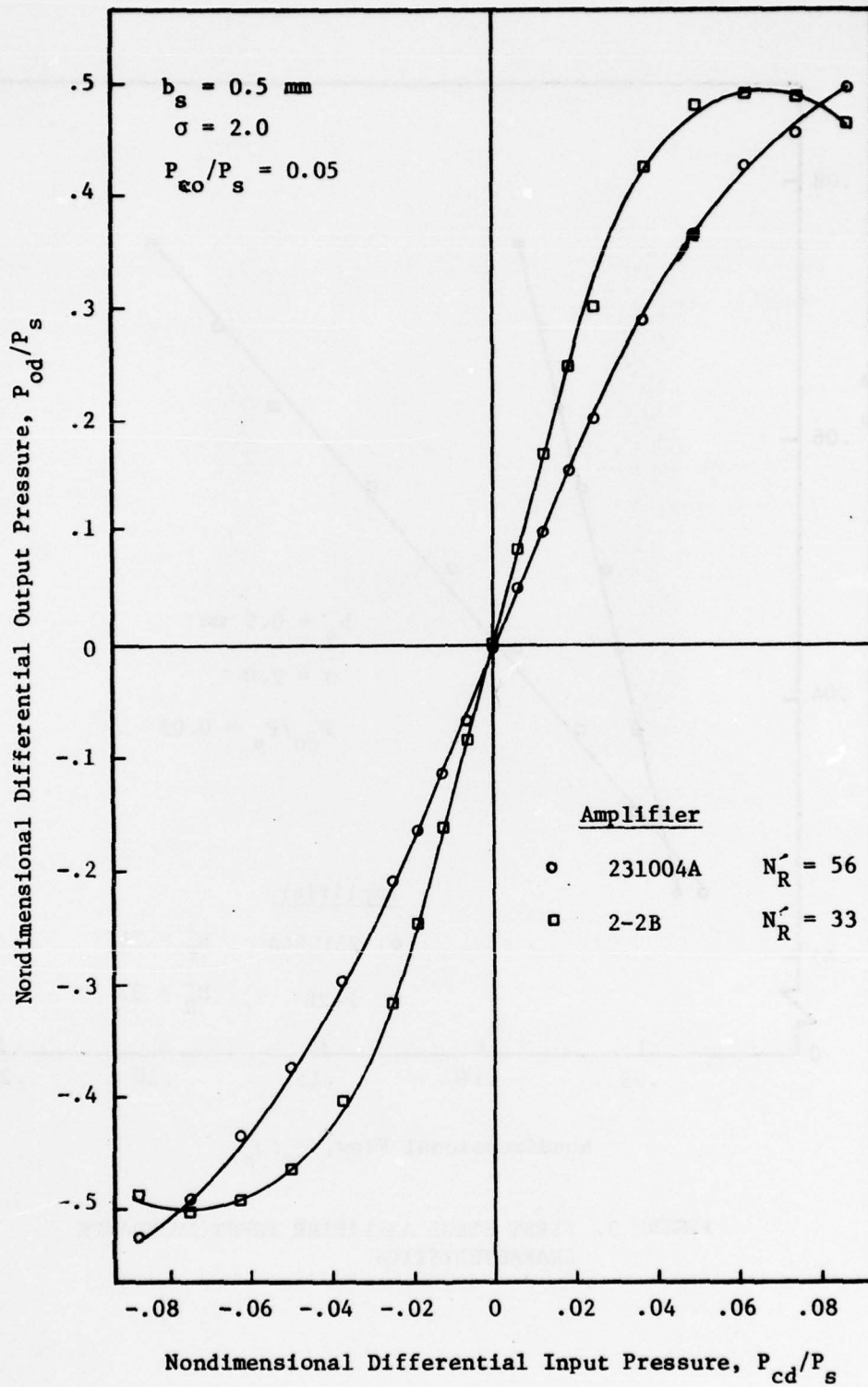


FIGURE 2. FIRST STAGE AMPLIFIER BLOCKED LOAD PRESSURE GAIN CHARACTERISTICS

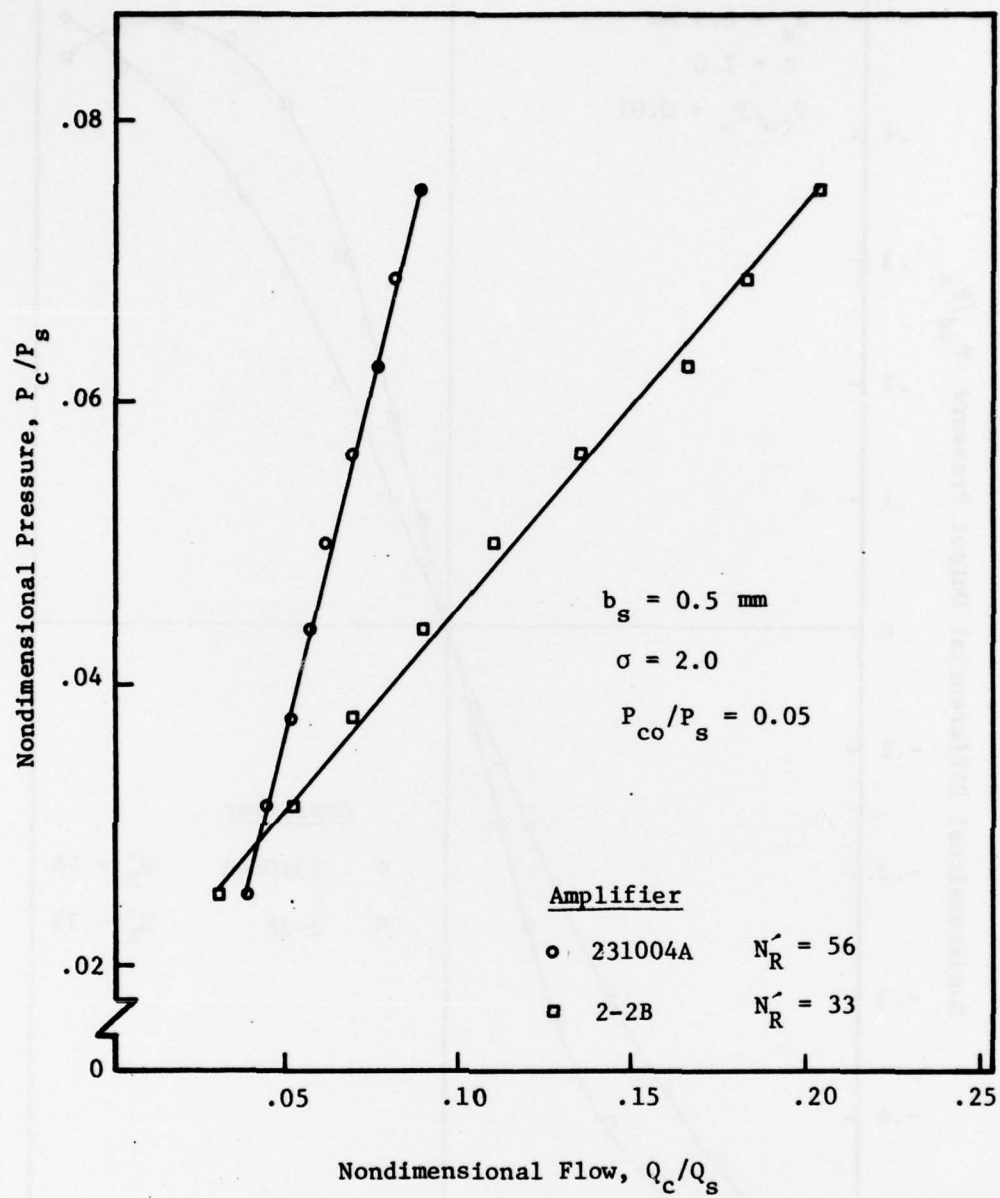


FIGURE 3. FIRST STAGE AMPLIFIER INPUT IMPEDANCE CHARACTERISTICS

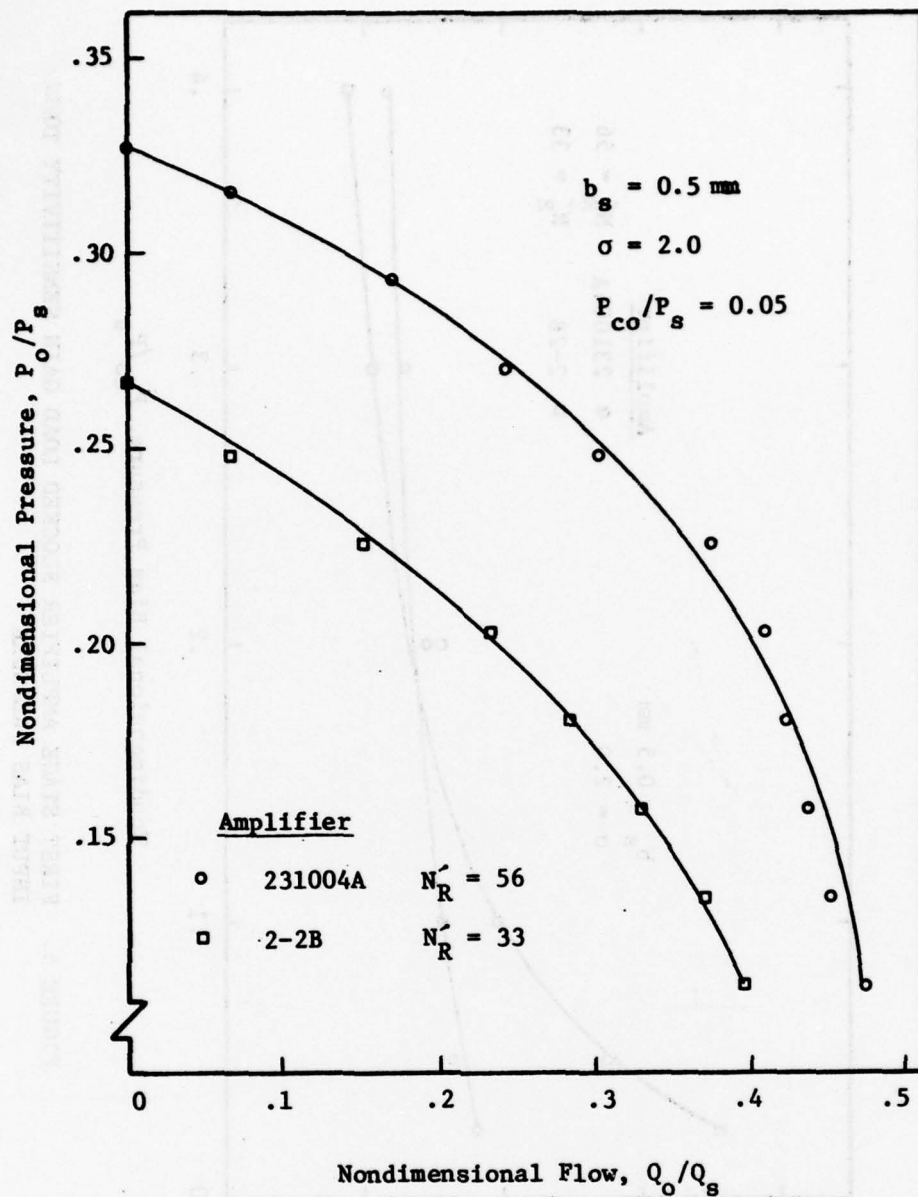


FIGURE 4. FIRST STAGE AMPLIFIER OUTPUT IMPEDANCE CHARACTERISTICS

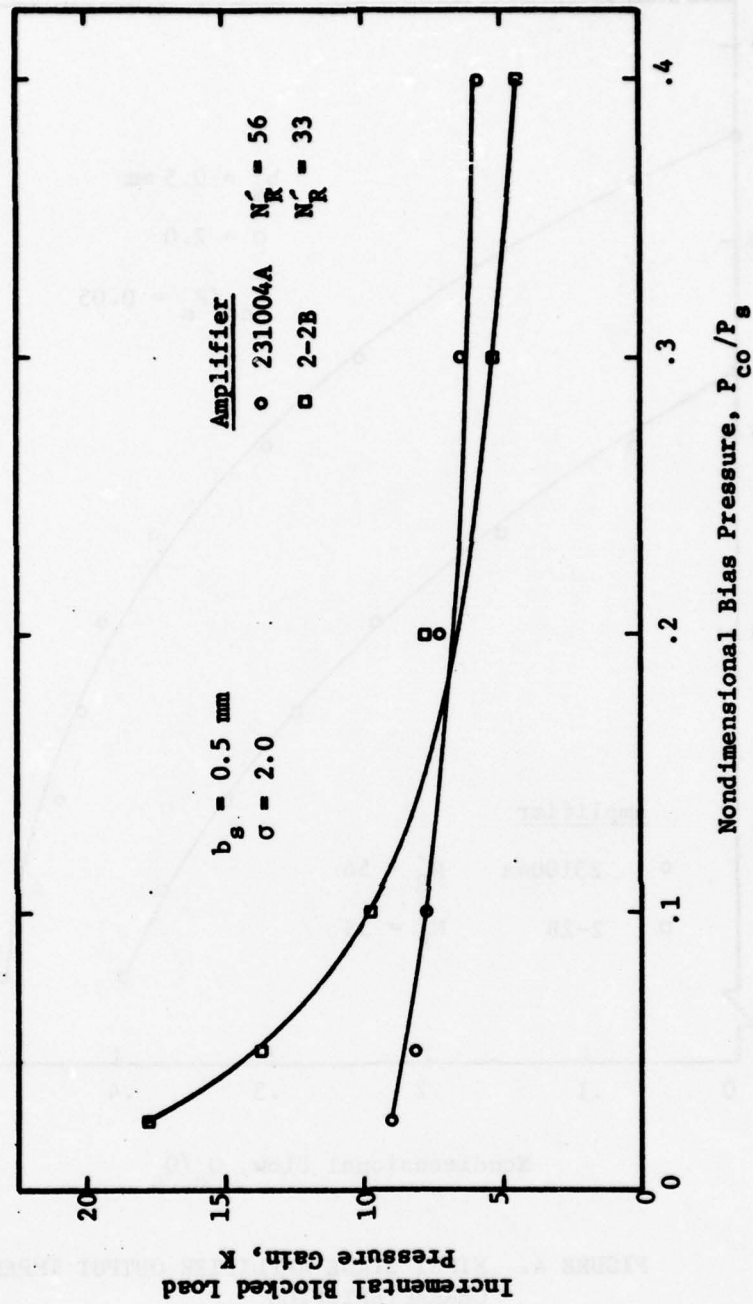


FIGURE 5. FIRST STAGE AMPLIFIER BLOCKED LOAD GAIN SENSITIVITY TO INPUT BIAS PRESSURE

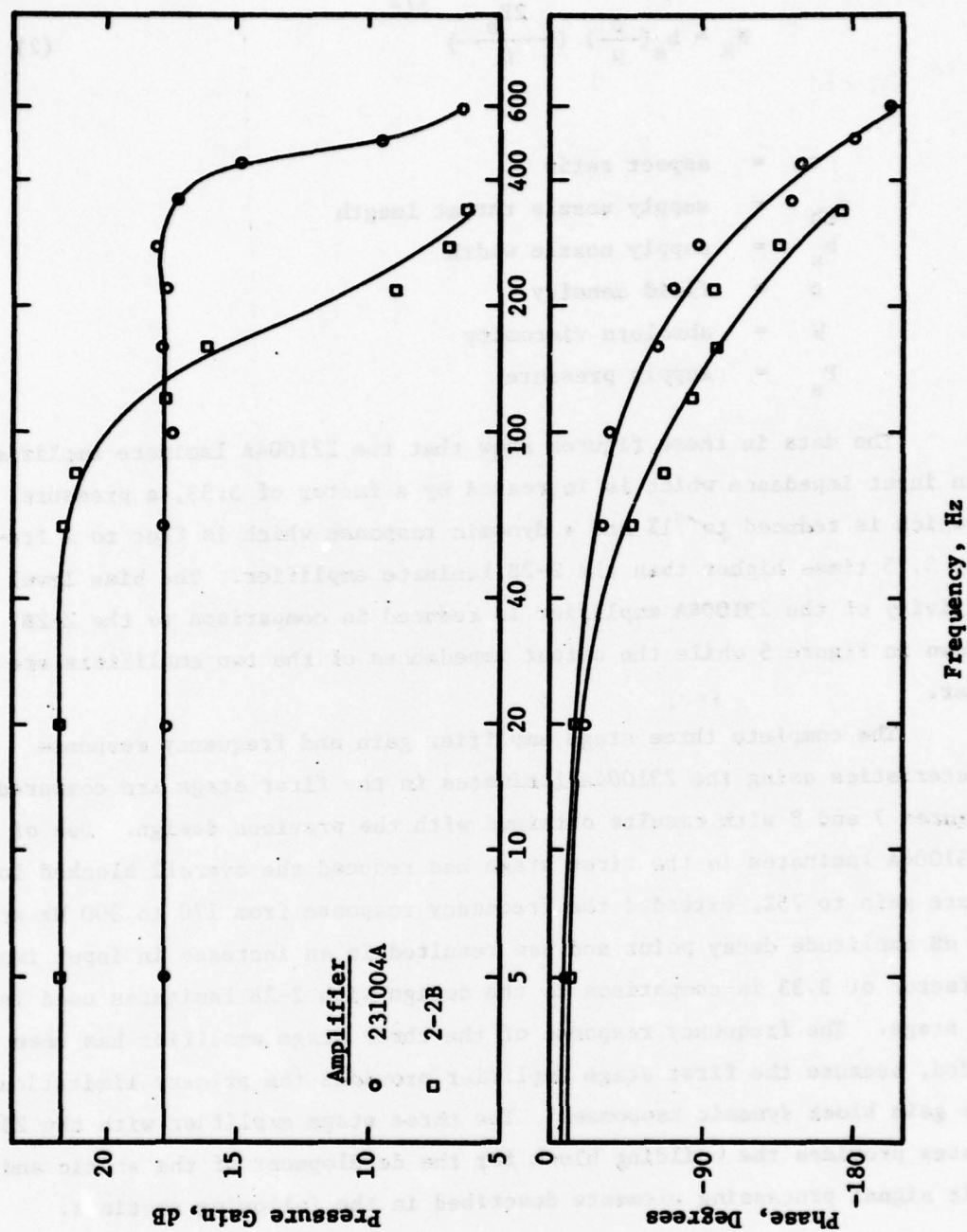


FIGURE 6. FIRST STAGE AMPLIFIER BLOCKED LOAD PRESSURE GAIN
FREQUENCY RESPONSE CHARACTERISTICS

where the Reynolds number N_R is defined as:

$$N_R = b_s \left(\frac{\rho}{\mu} \right) \left(\frac{2P_s}{\rho} \right)^{1/2} \quad (2)$$

and

- σ = aspect ratio
- x_{th} = supply nozzle throat length
- b_s = supply nozzle width
- ρ = fluid density
- μ = absolute viscosity
- P_s = supply pressure

The data in these figures show that the 231004A laminate amplifier has an input impedance which is increased by a factor of 3.33, a pressure gain which is reduced to 71% and a dynamic response which is flat to a frequency 3.75 times higher than the 2-2B laminate amplifier. The bias level sensitivity of the 231004A amplifier is reduced in comparison to the 2-2B as shown in Figure 5 while the output impedances of the two amplifiers are similar.

The complete three stage amplifier gain and frequency response characteristics using the 231004A laminates in the first stage are compared in Figures 7 and 8 with results obtained with the previous design. Use of the 231004A laminates in the first stage has reduced the overall blocked load pressure gain to 75%, extended the frequency response from 170 to 300 Hz at the 3 dB amplitude decay point and has resulted in an increase in input impedance by a factor of 3.33 in comparison to the design with 2-2B laminates used in the first stage. The frequency response of the three stage amplifier has been extended, because the first stage amplifier provides the primary limitation to the gain block dynamic response.¹ The three stage amplifier with the 231004A laminates provides the building block for the development of the static and dynamic signal processing elements described in the following sections.

¹Wormley, D.N. and Wilson, D.R., "Multistage Hydraulic Fluid Amplifier Characteristics," Technical Report HDL-TR-156-1, Harry Diamond Laboratories, 1974.

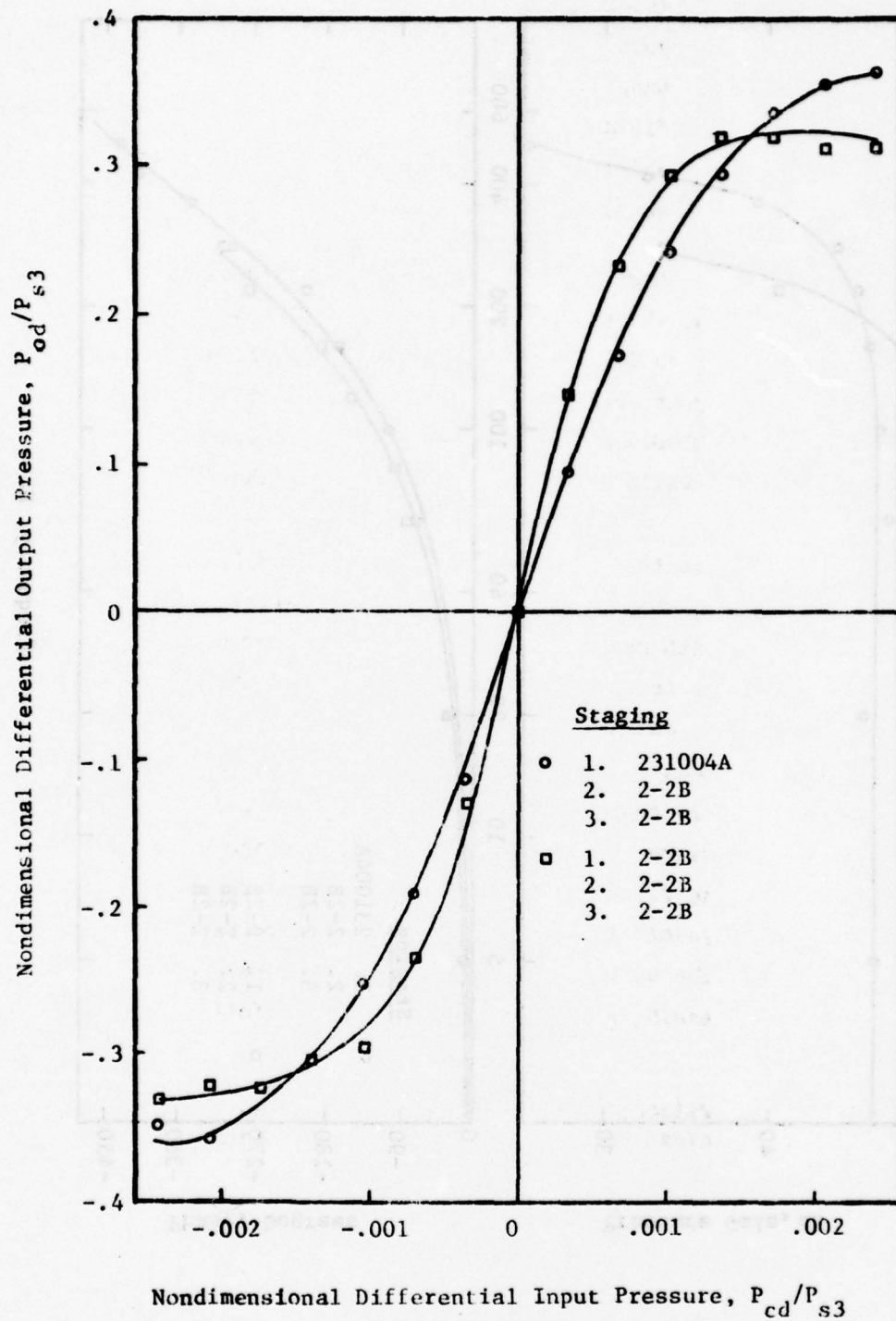


FIGURE 7. BLOCKED LOAD STATIC PRESSURE GAIN OF THREE STAGE AMPLIFIER

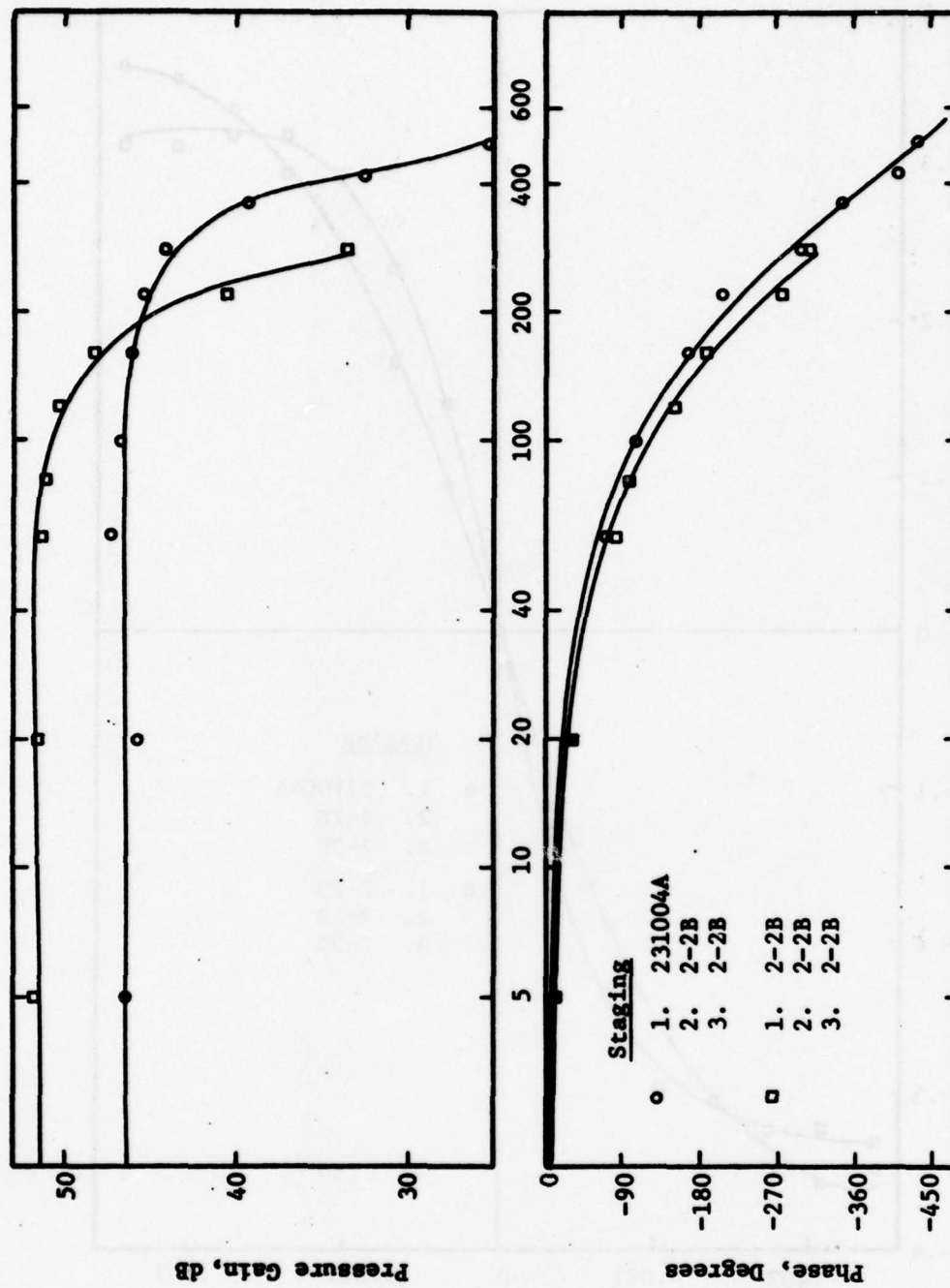


FIGURE 8. BLOCKED LOAD PRESSURE GAIN FREQUENCY RESONSE
OF THREE STAGE AMPLIFIER

To provide a basis for the analysis of circuits using the three stage amplifier, the amplifier dynamic output-input pressure transfer function has been modeled for small amplitude disturbances as a linear system which is represented as the product of a third order transfer function and a pure delay based upon an analysis of amplifier dynamics described by Wormley and Wilson.¹ The third order transfer function is represented as the product of a first order and a quadratic factor, while the pure delay is represented as a second order Padé expansion as shown below:

$$T(s) = \frac{P_{od}(s)}{P_{cd}(s)} = - \left[\frac{K}{T_1 s + 1} \right] \cdot \left[\frac{\omega_n^2}{s^2 + 2\xi\omega_n s + \omega_n^2} \right] \cdot \left[\frac{1 - 0.5\tau s + 0.125(\tau s)^2}{1 + 0.5\tau s + 0.125(\tau s)^2} \right] \quad (3)$$

where:

- s = Laplace Transform Operator
- P_{od} = gain block output differential pressure
- P_{cd} = gain block input differential pressure
- T = output-input pressure transfer function
- T_1 = first order lag time constant
- ω_n = second order factor natural frequency
- ξ = second order factor damping ratio
- K = static gain
- τ = delay time constant

The parameters in this transfer function were obtained by fitting this transfer function to measured data for the three stage gain block. The degree of fit is illustrated in Figure 9 for the following values of the constants:

$$\begin{aligned} K &= 215 \\ \omega_n &= 1634 \text{ rad/s} \\ \xi &= 0.6 \\ T_1 &= 6.12 \times 10^{-4} \text{ s} \\ \tau &= 1.33 \times 10^{-3} \text{ s} \end{aligned}$$

¹Wormley, D.N. and Wilson, D.R., "Multistage Hydraulic Fluid Amplifier Characteristics," Technical Report HDL-TR-156-1, Harry Diamond Laboratories, 1974.

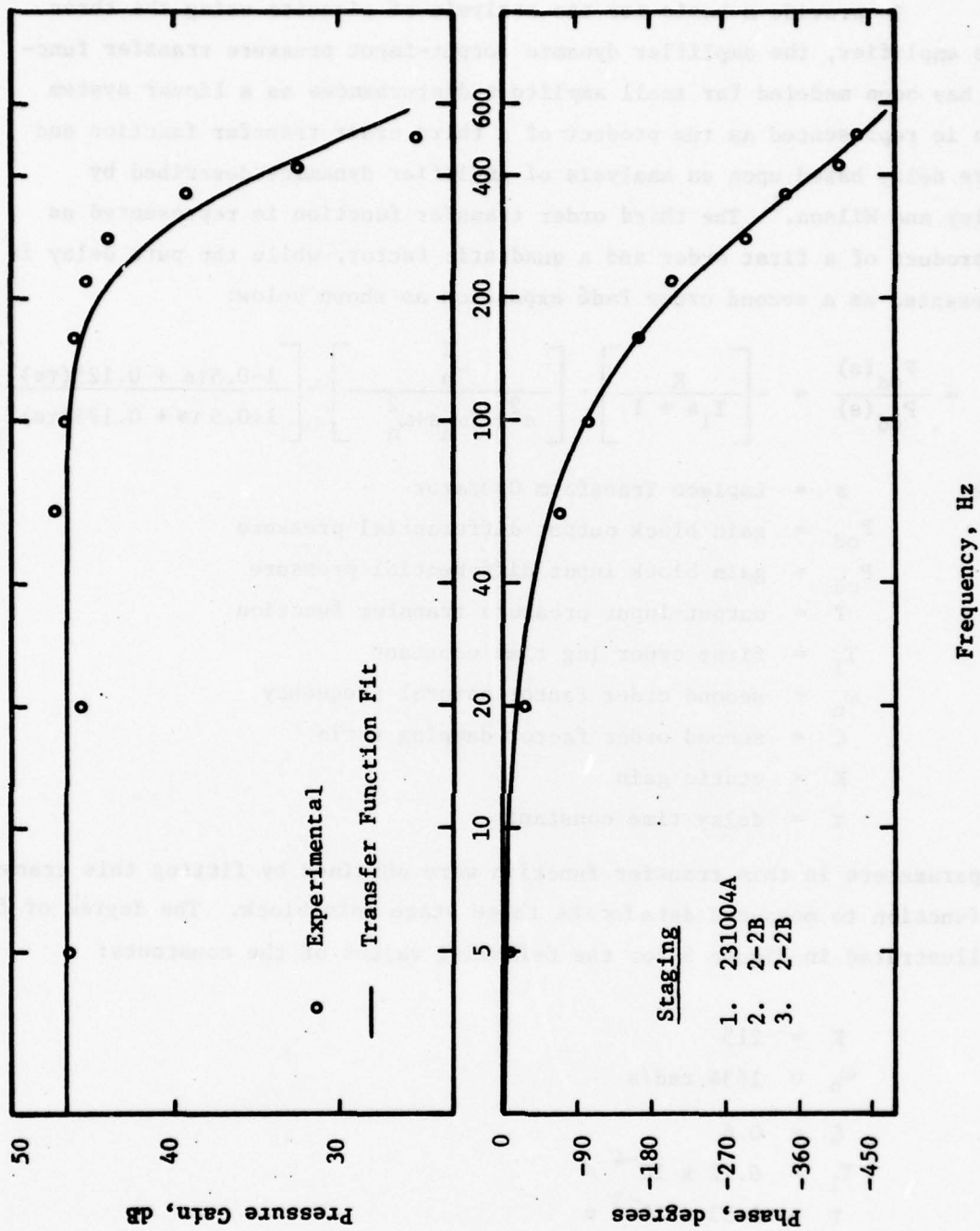


Figure 9. COMPARISON OF EXPERIMENTAL AND ANALYTICAL APPROXIMATION FOR THREE STAGE AMPLIFIER DYNAMIC RESPONSE

As shown in Figure 9, the agreement between the experimental data and the analytical representation is good. In the representation, the first and second order terms effectively represent the dynamics of the first stage amplifier, since its frequency response is the primary factor limiting the three stage amplifier; however, in addition to the delay time associated with the first stage, the additional delay times of the second and third stages contribute to τ . This delay term at 190 Hz contributes 90° phase shift to the amplifier frequency response.

A schematic circuit representation of the amplifier is illustrated in Figure 10 where the amplifier input impedance Z_a , transfer function $T(s)$ and output impedance Z_o are included in the representation.* This representation is used for preliminary design calculations in the static and dynamic signal processing circuits described below.

2.2 Summing Amplifier Characteristics

2.2.1 Circuit Model

A basic component of signal processing systems, summing amplifiers, can be constructed from the three stage gain block and passive fluid resistors. The general circuit shown in Figure 11 was considered as a basis for development of a fluid summing amplifier where each resistor is represented by an impedance Z , so that the effects of parasitic inertance in resistors can be included where appropriate.* The resistors Z_1 and Z_2 represent input resistors and Z_f represents a feedback resistor. The resistor Z_g has been included in the generalized model so that the nondimensional input bias pressure level may be adjusted to 5% at the amplifier, while the resistors Z_{D1} and Z_{D2} have been provided so that both feedback pressure levels and bias levels may be adjusted. When the resistors Z_g and Z_{D2} are infinite and Z_{D1} is zero, the circuit reduces to the standard summing amplifier circuit commonly used in electronic systems.

For the circuit as configured, the relationship between the differential output pressure P_{od} and the differential input pressures, P_{1d} and P_{2d} can be derived as:

*The circuit representation is a differential representation which shows half the actual number of physical components. Signals are shown as differential signals rather than single sided signals.

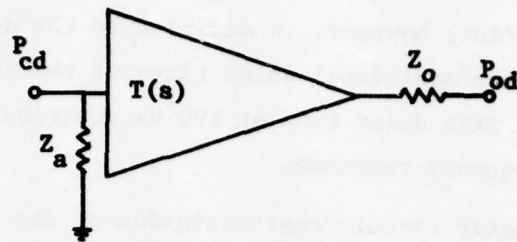


FIGURE 10. CIRCUIT SCHEMATIC OF THREE STAGE GAIN BLOCK

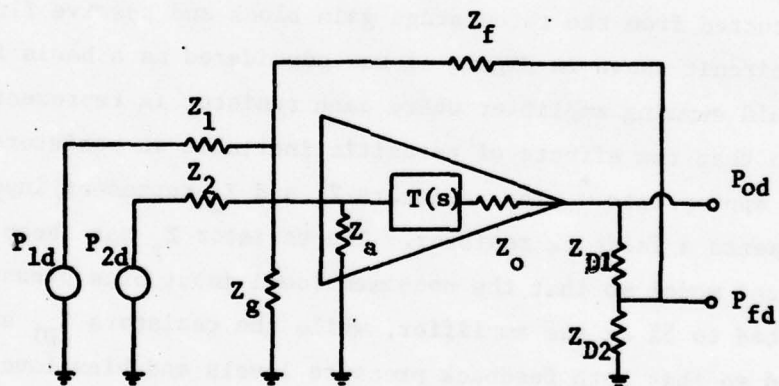


FIGURE 11. SUMMING AMPLIFIER SCHEMATIC

$$P_{od} = - \left[\frac{Z_f}{Z_1} P_{1d} + \frac{Z_f}{Z_2} P_{2d} \right] \cdot \left[\frac{T(s) R^*}{\frac{Z_f Z_o}{Z_{D1}} R^{**} \left(\frac{1}{Z_{D1}} + \frac{Z_o}{Z_f T Z_{D0}} \right) - \frac{Z_f Z_o}{Z_{D0}} + R^* R^{**} \left(\frac{T}{Z_{D1}} + \frac{Z_o}{Z_f Z_{D0}} \right)} \right] \quad (4)$$

$$\text{where: } \frac{1}{R^*} = \frac{1}{Z_g} + \frac{1}{Z_a} + \frac{1}{Z_1} + \frac{1}{Z_2} + \frac{1}{Z_f}$$

$$\frac{1}{R^{**}} = \frac{1}{Z_{D2}} + \frac{1}{Z_{D1}} + \frac{1}{Z_f} + \frac{Z_o}{Z_f T(s) Z_{D1}}$$

$$\frac{1}{Z_{D0}} = \frac{1}{Z_{D1}} + \frac{1}{Z_o}$$

The first term in brackets shows that the output pressure is simply a sum of $[Z_f/Z_1]P_{1d}$ and $[Z_f/Z_2]P_{2d}$ as occurs in a classical summing amplifier. The second term in brackets reflects the effects of amplifier dynamics, $T(s)$, finite input impedance Z_a and output impedance Z_o and of Z_g , Z_{D1} and Z_{D2} .

2.2.2 Resistor Characteristics

Input and feedback resistors were developed for the summing amplifier from small diameter stainless steel tubing. These resistor characteristics are summarized in Figure 12. The resistors have nearly linear characteristics for the range of pressures and flows considered. One resistor has a resistance $R = 1.5 \times 10^{10} \text{ N-s/m}^5$ which is approximately equal to the amplifier input resistance, while the second resistor has $R = 1.4 \times 10^{12} \text{ N-s/m}^5$ which is 93 times the value of the input resistor. These resistances were selected as target values for input and feedback resistors so that input pressures to the summing amplifier were in the 0 to 7 kPa (0-1 psi) range with a 5% nondimensionalized pressure bias at the amplifier input. For each resistor the parasitic inductance I_i was estimated from slug flow as:

$$I_i = \frac{\rho \ell_i}{A_i} \quad (5)$$

where: I_i = i^{th} resistor inductance
 A_i = i^{th} tube area
 ℓ_i = i^{th} tube length

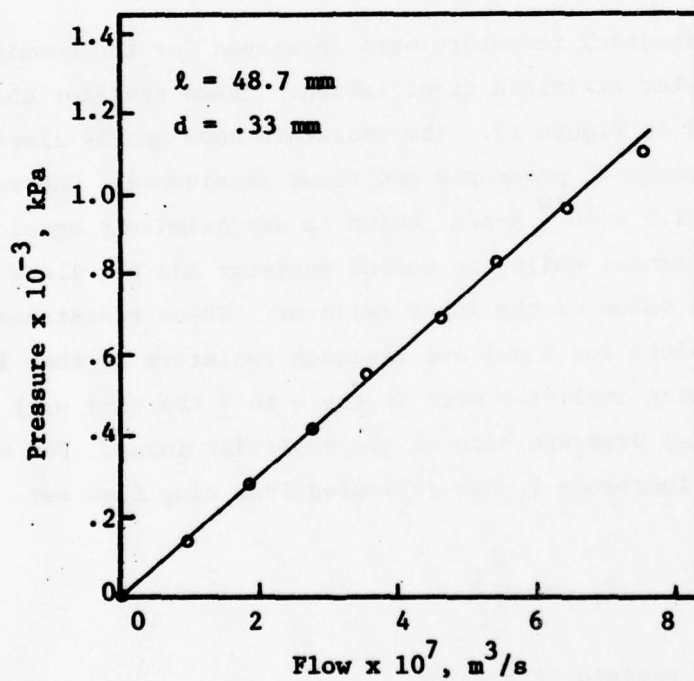
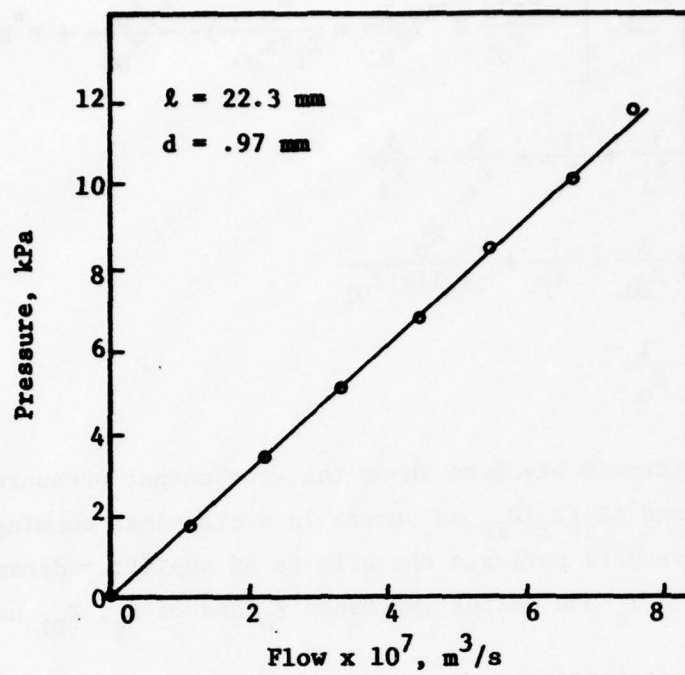


FIGURE 12. RESISTOR CHARACTERISTICS

The impedance of each resistor can then be written as:

$$Z_1 = R_1(\tau_1 s + 1)$$

where: $\tau_1 = I_1/R_1$ = resistor time constant

The time constants for each resistor is tabulated below:

$R = 1.5 \times 10^{10} \text{ N-s/m}^5$	$\tau = 1.78 \times 10^{-3} \text{ s}$
$R = 1.4 \times 10^{12} \text{ N-s/m}^5$	$\tau = 3.4 \times 10^{-4} \text{ s}$

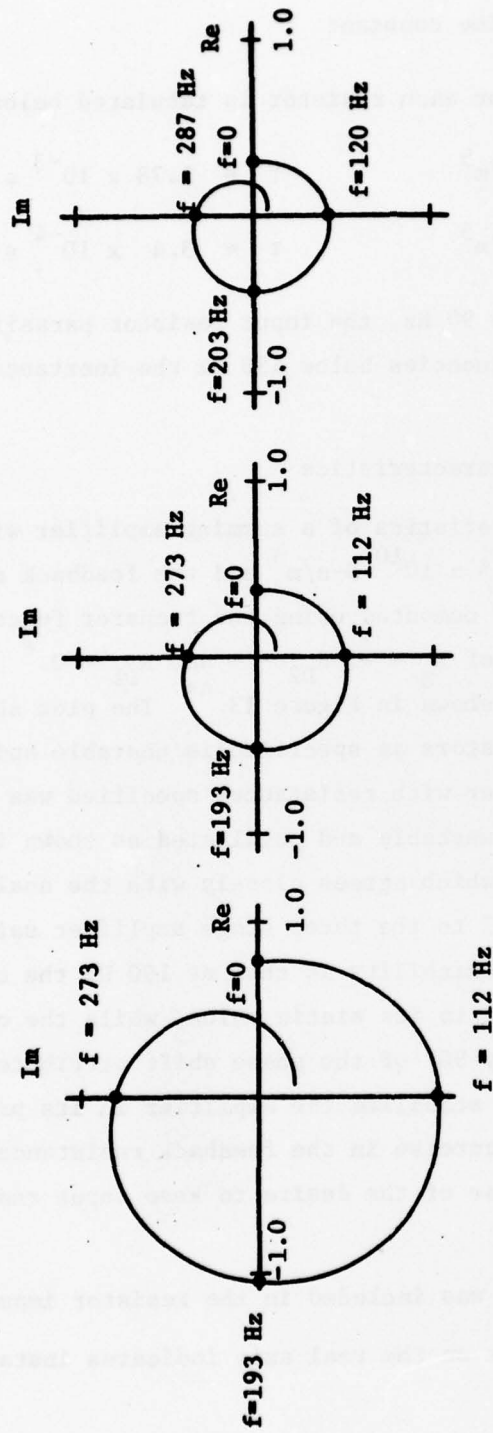
For frequencies below 90 Hz, the input resistor parasitic inductance may be neglected while for frequencies below 450 Hz the inductance of the feedback resistor may be neglected.

2.2.3 Amplifier Performance Characteristics

The stability characteristics of a summing amplifier with input resistances R_1 and R_2 equal to $1.5 \times 10^{10} \text{ N-s/m}^5$ and the feedback resistor R_f equal to $1.4 \times 10^{12} \text{ N-s/m}^5$ were computed using the transfer functions of equations (3) and (4) for the case of $R_g = \infty$, $R_{D2} = \infty$ and $R_{D1} = 0$.^{*} The resulting Nyquist plot for the system is shown in Figure 13.^{**} The plot shows that the summing amplifier with the resistors as specified is unstable and should oscillate at 193 Hz. The amplifier with resistances specified was constructed and tested. The amplifier was unstable and oscillated as shown in Figure 14 where the frequency is 189 Hz, which agrees closely with the analysis. The peak to peak pressures are equal to the three stage amplifier saturation level. The principal reason for this instability is that at 190 Hz the open loop gain of the amplifier is nearly equal to its static value, while the open loop phase shift is 180° with approximately 90° of the phase shift attributed to the pure delay term in equation (3). To stabilize the amplifier in its present configuration requires either an increase in the feedback resistance or a decrease in the input resistance. Because of the desire to keep input resistors at their

* The inductance of each resistor was included in the resistor impedances.

** Encirclement of the -1.0 point on the real axis indicates instability in the Nyquist plot.



(a) Unstable System with Base Feedback Resistor (b) Stable System with Pressure Divider (c) Stable System with Base Feedback Resistor Doubled

FIGURE 13. NYQUIST STABILITY PLOT OF SUMMING AMPLIFIER

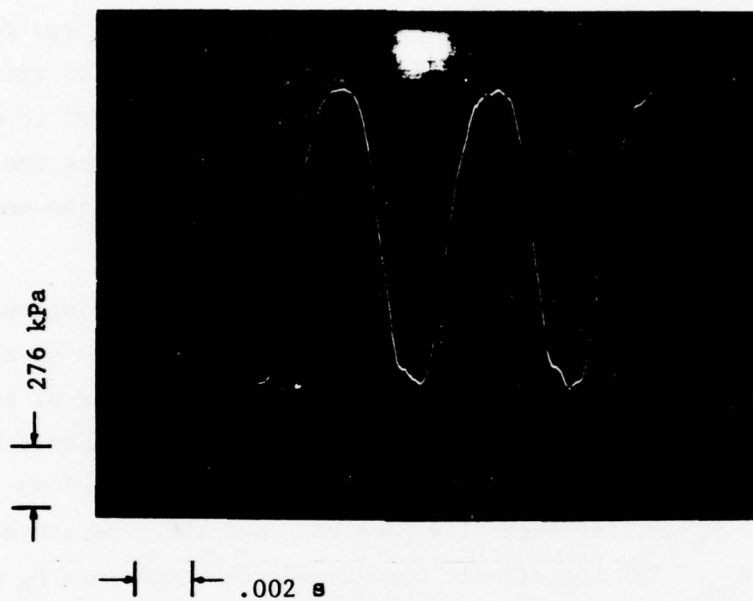


FIGURE 14. SUMMING AMPLIFIER OUTPUT PRESSURE
OSCILLATION

design nominal values so that input pressure levels could be maintained in the 0 to 7 kPa (1-0 psi) range, altering the feedback resistance was considered primarily. A Nyquist plot, computed using equation (4), with a capillary resistor whose value is twice that of the base resistor is presented in Figure 13.* This data shows that a resistor that is twice the length of the base resistor will stabilize the feedback amplifier, or alternately a resistor which is seventy per cent of the area of the original resistor could be used. While increasing the length or decreasing the area of the feedback resistor could provide a stable amplifier, neither alternative is attractive. Increasing the resistor length yields a tube of sufficient length to be inconvenient in maintaining a small, compact unit, while decreasing the area further results in a tube diameter which is less than one half the smallest flow dimension associated with the amplifiers.

Rather than increasing the resistor length or decreasing the area, an alternative method of stabilizing the amplifier has been used as shown in Figure 11, where the feedback path has been modified with a pair of resistors R_{D1} and R_{D2} that act as a pressure divider. For the case considered $R_{D1} = R_{D2} = R_D$. When $R_D \gg R_O$, the pressure divider circuit does not load the amplifier and when $R_f \gg R_D$ the feedback flow does not load the pressure divider so that $P_{fd} = 1/2 P_{od}$. For the circuit considered, the resistors R_D were constructed from 0.66 mm diameter orifices so that their diameters were comparable to the smallest amplifier dimension. For these orifices, the ratios $R_D/R_O = 7.0$ and $R_f/R_D = 11.4$, thus the orifices provide a nearly ideal pressure divider. With the feedback pressure $P_{fd} = 1/2 P_{od}$, the feedback circuit should be stable as shown in the Nyquist plot of Figure 13.

The amplifier was set up in the configuration with the pressure divider as shown in the schematic and photograph of Figure 15. For this configuration, the static pressure gain for a single differential input to the summing amplifier with the second input grounded is simply a function of static resistances and from equation (4) can be calculated as 63. Test results run on the amplifier are shown in Figure 16. The data illustrate a linear gain of 60 with a null offset less than 5% of full scale value.

* Both the resistance and inductance increase linearly with length.

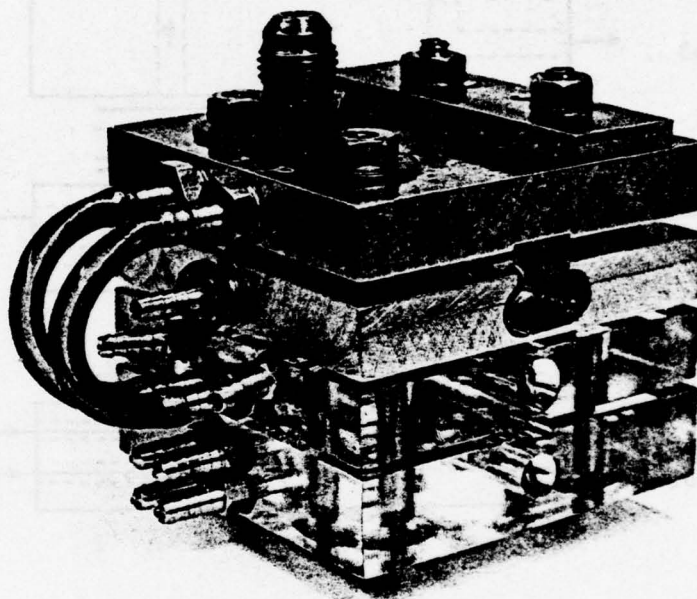


FIGURE 15a. PHOTOGRAPH OF SUMMING AMPLIFIER

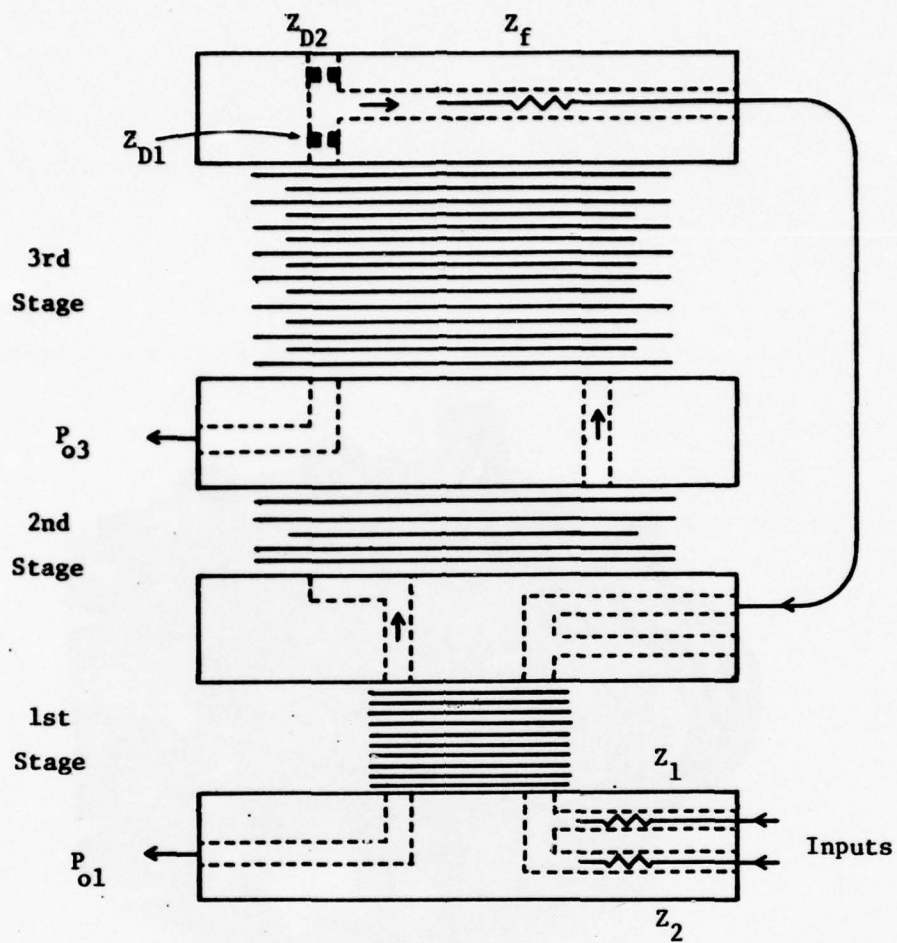


FIGURE 15b. SCHEMATIC OF SUMMING AMPLIFIER

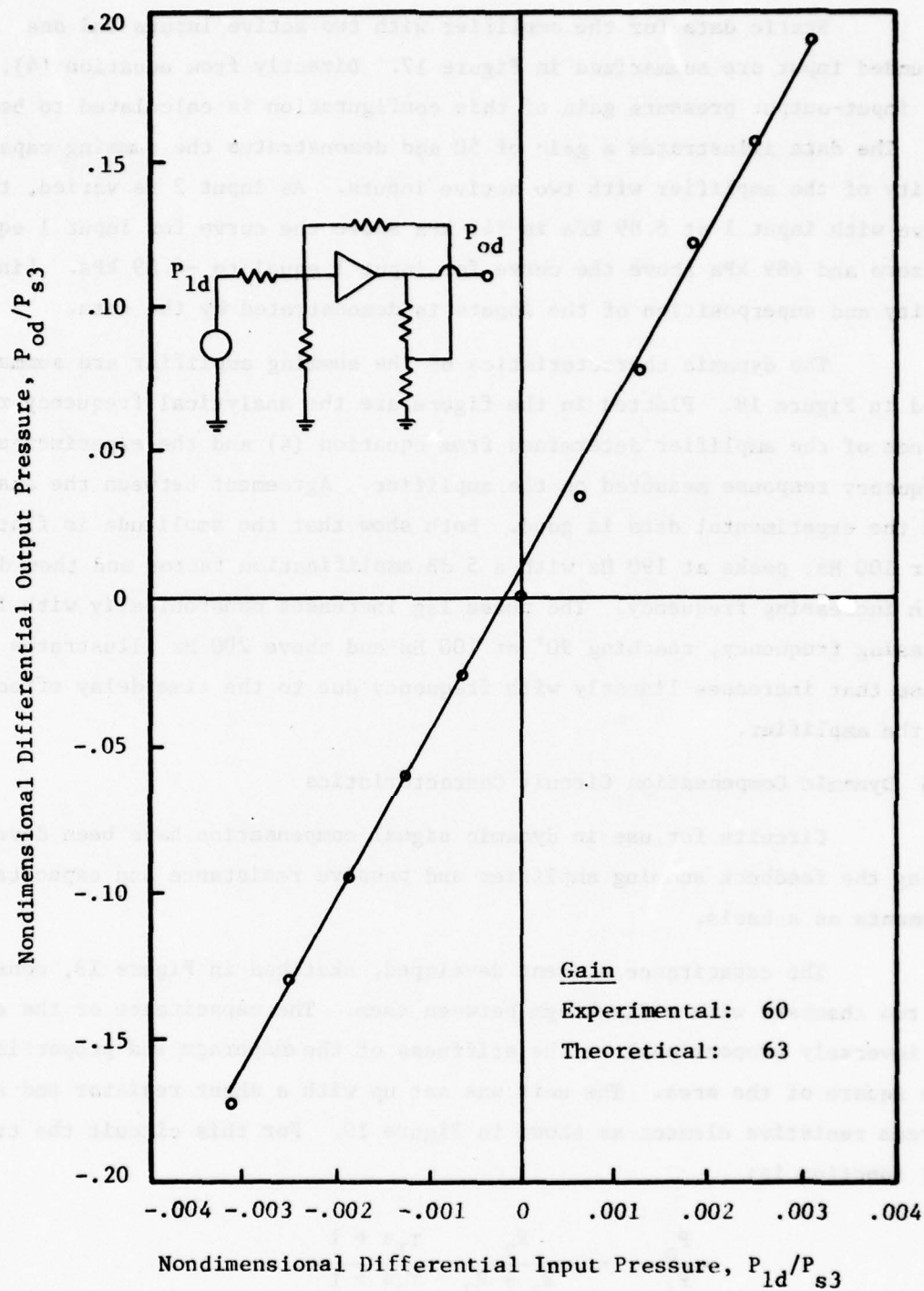


FIGURE 16. SUMMING AMPLIFIER SINGLE-INPUT STATIC GAIN CHARACTERISTICS

Static data for the amplifier with two active inputs and one grounded input are summarized in Figure 17. Directly from equation (4), the input-output pressure gain of this configuration is calculated to be 49. The data illustrates a gain of 50 and demonstrates the summing capability of the amplifier with two active inputs. As input 2 is varied, the curve with input 1 at 6.89 kPa is 345 kPa above the curve for input 1 equal to zero and 689 kPa above the curve for input 1 equal to -6.89 kPa. Linearity and superposition of the inputs is demonstrated by the data.

The dynamic characteristics of the summing amplifier are summarized in Figure 18. Plotted in the figure are the analytical frequency response of the amplifier determined from equation (4) and the experimental frequency response measured on the amplifier. Agreement between the analysis and the experimental data is good. Both show that the amplitude is flat to over 100 Hz, peaks at 190 Hz with a 5 dB amplification factor and then decreases with increasing frequency. The phase lag increases monotonically with increasing frequency, reaching 90° at 100 Hz and above 200 Hz illustrates a phase that increases linearly with frequency due to the time delay effects in the amplifier.

2.3 Dynamic Compensation Circuit Characteristics

Circuits for use in dynamic signal compensation have been developed using the feedback summing amplifier and passive resistance and capacitance elements as a basis.

The capacitance element developed, sketched in Figure 19, consists of two chambers with a diaphragm between them. The capacitance of the element is inversely proportional to the stiffness of the diaphragm and proportional to the square of the area. The unit was set up with a shunt resistor and a downstream resistive element as shown in Figure 19. For this circuit the transfer function is:

$$\frac{P_o}{P_i} = \frac{R_2}{R_1 + R_2} \frac{\tau_1 s + 1}{\tau_2 s + 1} \quad (6)$$

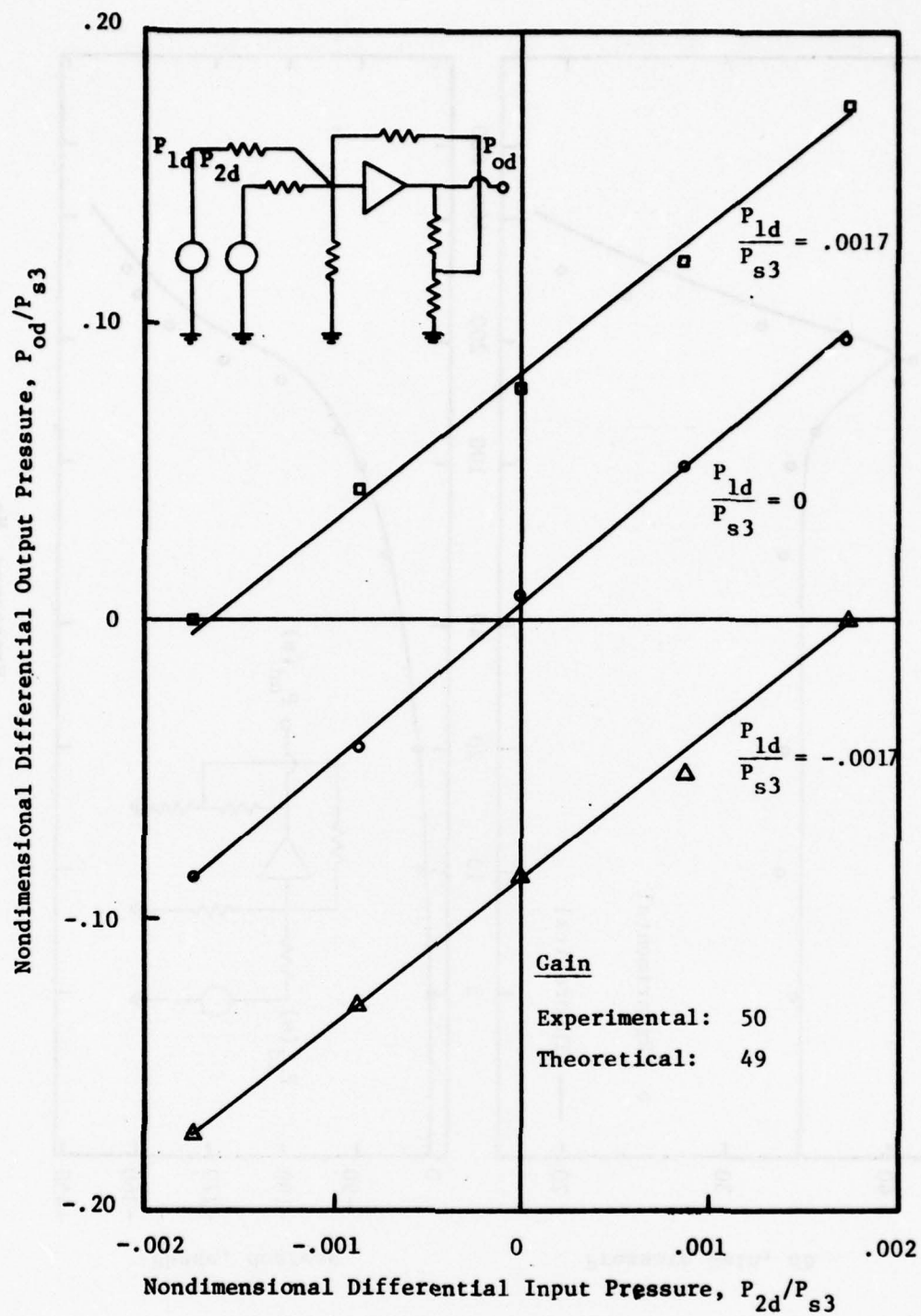


FIGURE 17. SUMMING AMPLIFIER MULTI-INPUT STATIC GAIN CHARACTERISTICS

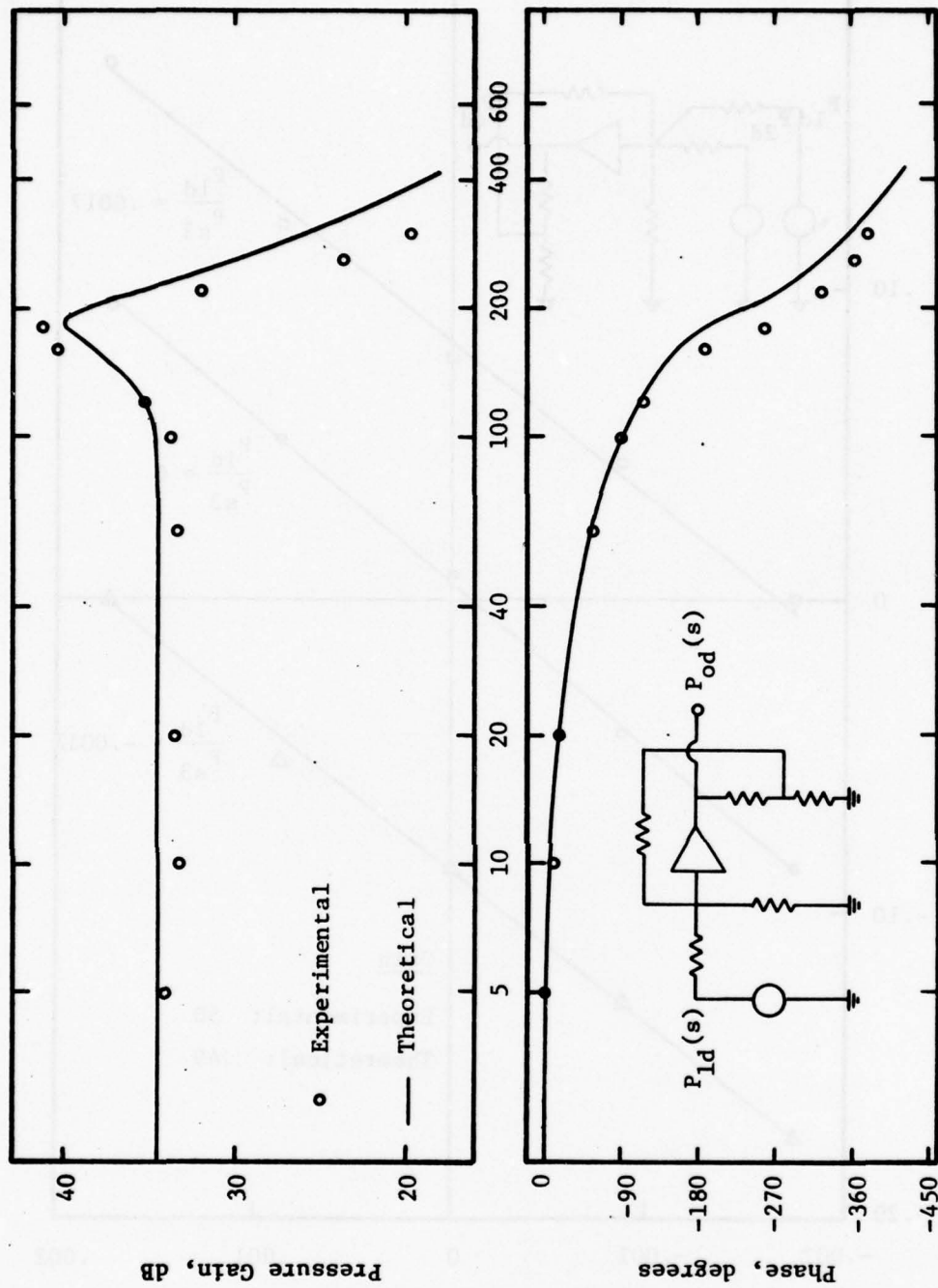


FIGURE 18. BLOCKED LOAD FREQUENCY RESPONSE OF SUMMING AMPLIFIER

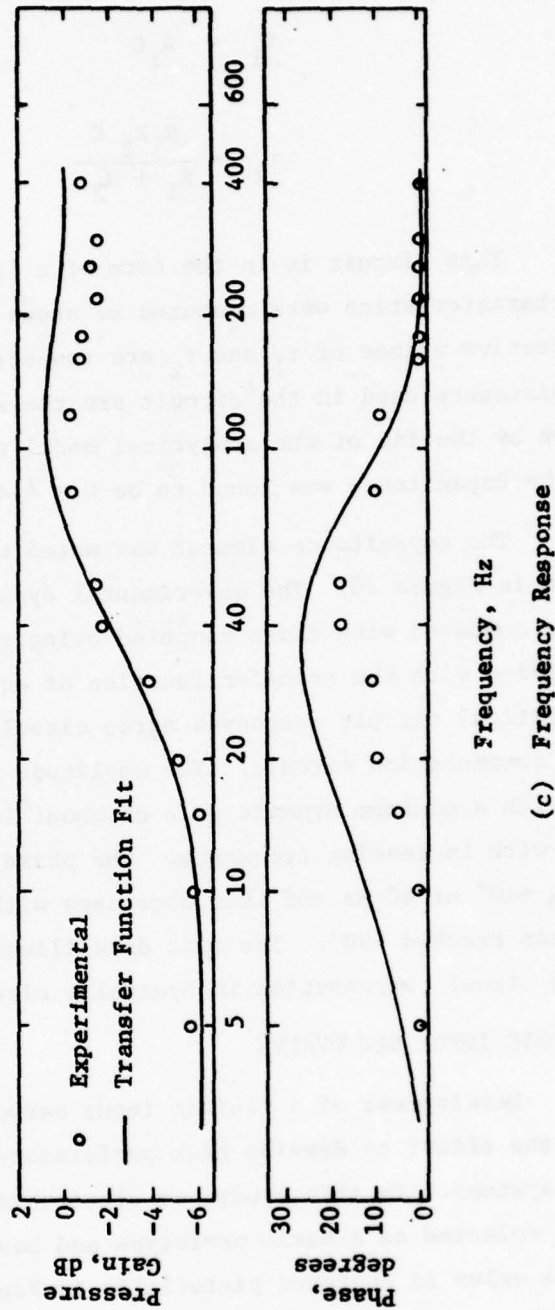
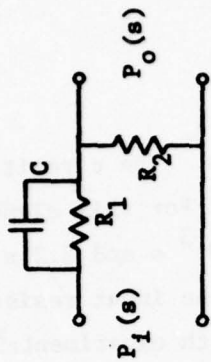
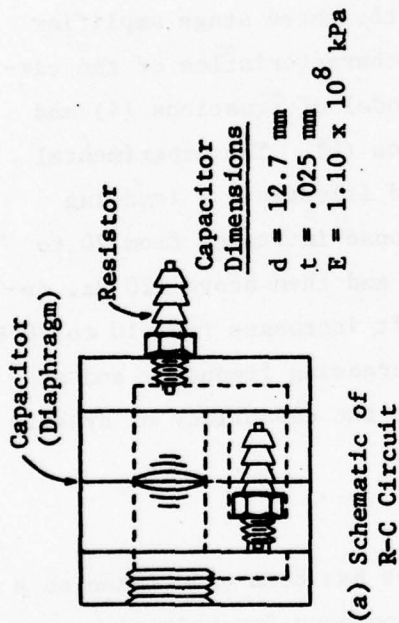


FIGURE 19. SKETCH AND DYNAMIC CHARACTERISTICS OF FLUID RESISTOR-CAPACITOR ELEMENT

where:

$$\tau_1 = R_1 C$$

$$\tau_2 = \frac{R_1 R_2 C}{R_1 + R_2}$$

This circuit is in the form of a lead-lag circuit. The circuit dynamic characteristics were measured as shown in Figure 19. For this element the effective values of τ_1 and τ_2 are respectively 6.4×10^{-3} s and 3.2×10^{-3} s. The resistances used in the circuit are the same value as the input resistors. As shown by the fit of the analytical model equation (6) with experimental data, the capacitance was found to be $C = 4.4 \times 10^{-13}$ m⁵/N.*

The capacitance element was mated with the three stage amplifier as shown in Figure 20. The experimental dynamic characteristics of the circuit are compared with those computed using the model of equations (4) and (5) modified with the transfer function of equation (6). The experimental and analytical circuit responses agree closely and illustrate a lead-lag type of compensation circuit. The amplitude response increases from 20 to 100 Hz with a maximum dynamic gain of about 10 db and then above 120 Hz, decreases with increasing frequency. The phase shift increases from 10 to 40 Hz reaching +10° at 40 Hz and then decreases with increasing frequency and at 100 Hz has reached -90°. The test data illustrate the capability of dynamic lead-lag signal compensation in hydraulic circuits.

3. FLUIDIC INPUT SERVOVALVE

Development of a fluidic input servovalve has been undertaken as a part of the effort to develop high performance components for hydraulic actuation systems. In this study, an electrohydraulic two stage servovalve has been selected as a basic prototype and has been modified for fluidic input. The valve is sketched pictorially in Figure 21 with the fluidic input modifications. The valve is a Moog Model 50 x 451 servovalve** which has

*The capacitance was determined experimentally by the data of Figure 19, and equation (6), since computation of the effective diaphragm stiffness in its support structure could not be performed accurately.

**Although not shown in the schematic, the valve also has capability for direct mechanical input through an arm that moves a sleeve encompassing the spool. This mechanical input was not used in the tests described in this study.

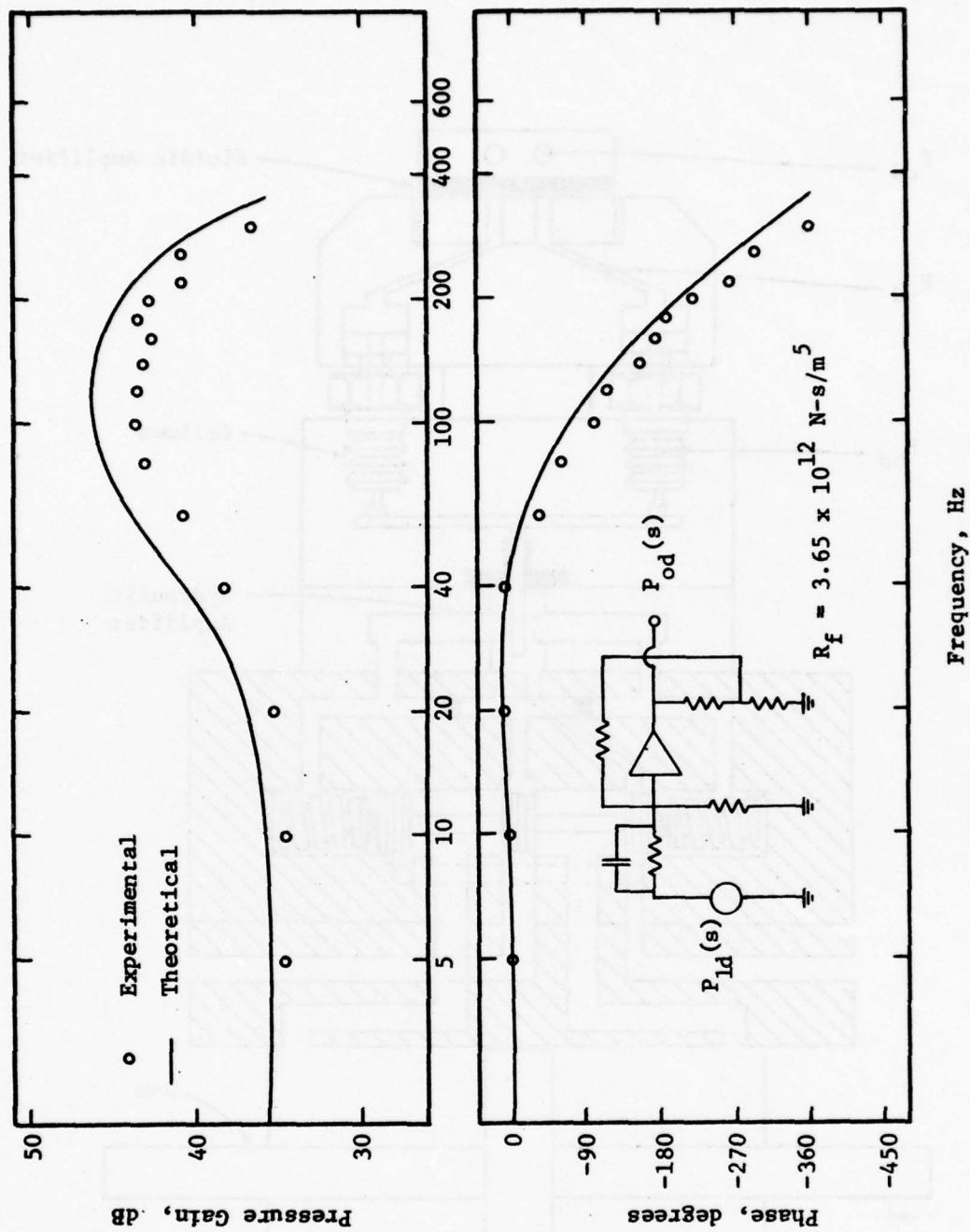


FIGURE 20. FREQUENCY RESPONSE CHARACTERISTICS OF DYNAMIC COMPENSATION CIRCUIT

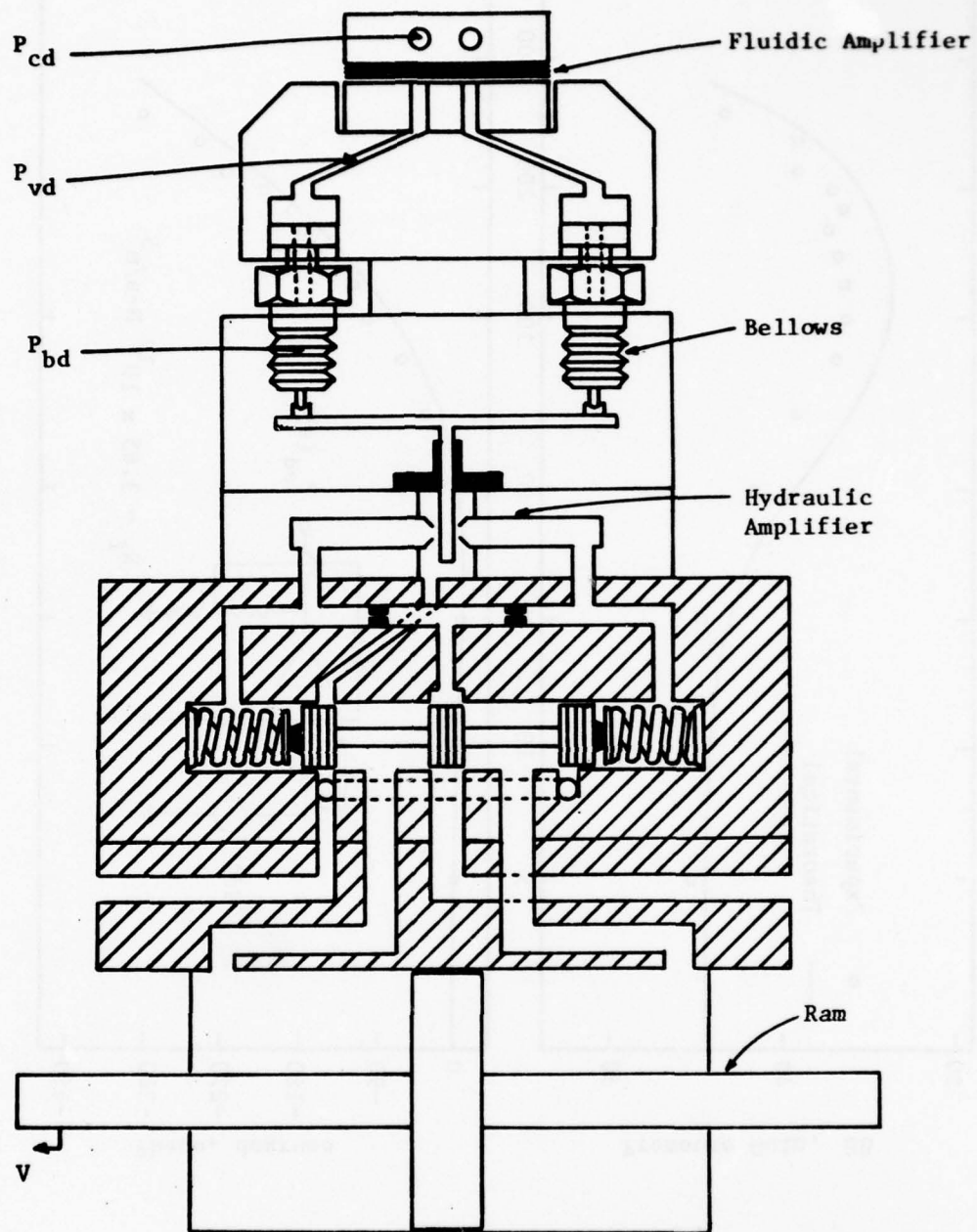


FIGURE 21. SCHEMATIC OF TWO STAGE SERVOVALVE

had the torque motor deactivated and has had bellows installed in order to position the first stage flapper. The valve consists of a pair of bellows which receive low pressure fluid inputs. The bellows provide a force differential on the flapper. The flapper controls the openings in the first stage flapper nozzle valve which in turn provides a pressure differential to the ends of the second stage spool. In the fluidic input valve the force balance involves the bellows forces and the flapper forces; thus, the only modification from the commercial version of the valve, is replacement of the torque motor forces on the flapper with the bellows forces. The commercial electrohydraulic valve characteristics and the bellows dimensions are summarized in Table II.

The valve with bellows installed was first tested statically. A differential pressure was applied to the bellows and then the pressure differential developed by the first stage flapper-nozzle across the spool ends and the flow from the second stage were measured with no load across the second stage output. The first stage pressure data is presented in Figure 22 while the second stage flow data is presented in Figure 23. The first stage differential pressure gain per unit differential bellows pressure data illustrates good linearity with a differential gain of 44. The second stage output flow per unit differential bellows pressure data is linear with a gain of $5.0 \times 10^{-10} \text{ m}^5/\text{N-s}$.

To provide the servovalve with low level fluidic input capability of 7 kPa, a fluid amplifier input stage was designed to drive the bellows assembly. The design of the fluid amplifier was based upon a single stage amplifier consisting of 231004A laminates. The design goals for the amplifier were to provide a pressure gain greater than 5, with input levels of 0-7 kPa so that the bellows can be driven to its full range. Also a primary consideration in the design is dynamic performance. It is desired to have the fluidically driven valve dynamic performance be comparable to that of the electrohydraulic commercial valve. A single stage amplifier section was constructed with characteristics as summarized in Table III. This section is similar to the first stage amplifier of the three stage gain block and thus could be expected to provide the static gain characteristics desired. To prevent dynamic loading of the amplifier due to the bellows capacitance-amplifier output impedance, the time constant associated with the bellows

TABLE II. SERVOVALVE BELLOWS PARAMETERS

Control Pressure Ratings

	<u>Rated</u>	<u>Proof</u>
Differential Input	34.5 kPa (5 psid)	68.9 kPa (10 psid)
Input High Side, with Respect to Return		276 kPa (40 psid)
Inputs and Return, in Common		689.5 kPa (100 psig)

Servo Valve Pressure Rating

	<u>Rated</u>	<u>Proof</u>
Supply	1.0×10^4 kPa (1500 psi)	2.06×10^4 kPa (3000 psi)
Supply and Return in Common		1.0×10^4 kPa (1500 psi)

Bellows Effective Area

31.9 mm^2
(.0495 in²)

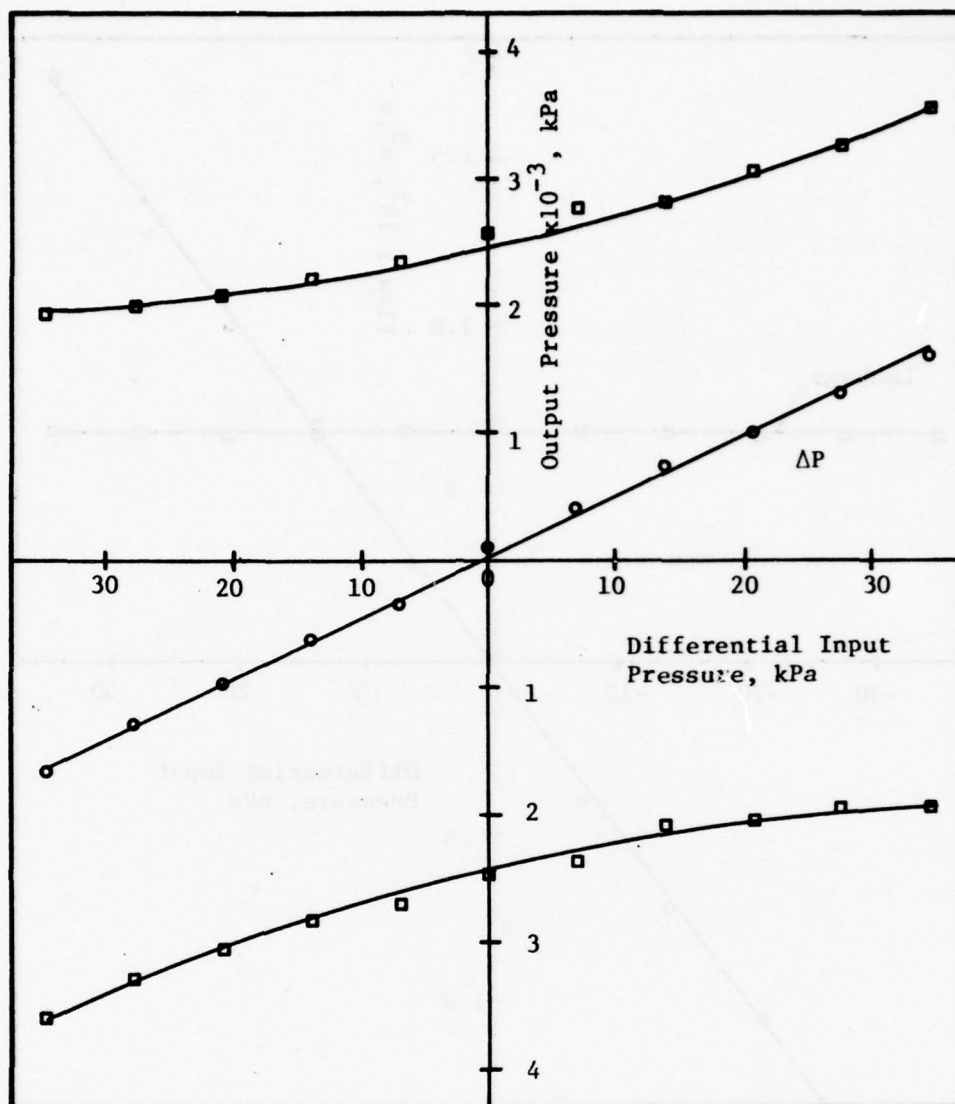


FIGURE 22. VALVE FIRST STAGE PRESSURE GAIN

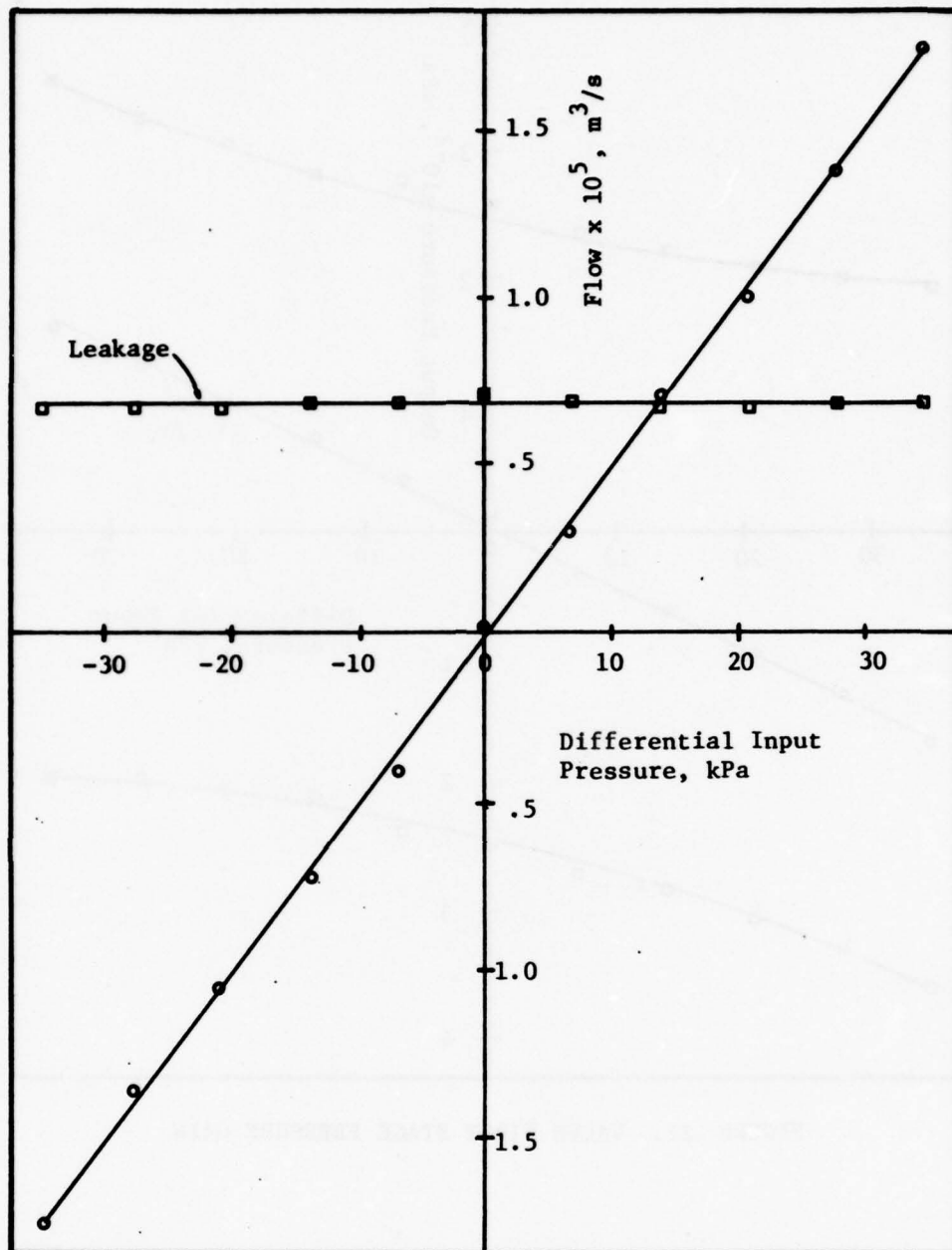


FIGURE 23. VALVE FLOW VERSUS INPUT TO BELLOWS

TABLE III. SINGLE STAGE AMPLIFIER PARAMETERS

Laminate Design	231004A
Size, b_s (mm)	0.5
Aspect Ratio, σ	2
Supply Pressure, P_s (kPa) [psi]	110 [16]
Nondimensional Bias Pressure, P_{co}/P_s	.05
Static Pressure Gain	7

TABLE IV. RAM CHARACTERISTICS

Effective Piston Area	435. mm ²
Piston Mass	.078 kg

capacitance - amplifier output impedance was reduced by using two sections in parallel, thus cutting the output impedance by a factor of two. In exploratory dynamic testing, when only one section was used, distortion occurred in the bellows pressure signal.

The complete fluid amplifier-valve assembly was tested first statically and then dynamically. The static test data are summarized in Figure 24 and 25 as first stage flapper-nozzle valve output pressure versus fluid amplifier input pressure and as second stage valve output flow under no load versus fluid amplifier input pressure. Both sets of data show a good degree of linearity. The first stage pressure gain with respect to the fluid amplifier input is 320 and at a fluid amplifier input pressure of 6 kPa the full bellows pressure is achieved. The valve output flow to the fluid amplifier input pressure gain is $3.3 \times 10^{-9} \text{ m}^3/\text{N-s}$. Thus the static data demonstrate that with input pressures in the range of 0 to 7 kPa, full valve operation may be achieved.

Dynamic data were taken with the valve output connected to a small hydraulic ram. The ram characteristics are summarized in Table IV. With this light weight ram, the ram velocity is a measure of the valve flow under no load conditions. Tests were run by introducing a sinusoidally varying pressure signal at the fluid amplifier input and then measuring this input pressure, the pressure at the bellows and the ram velocity. The pressures were measured with piezoelectric pressure transducers while the ram velocity was measured with a LVSYN velocity transducer. The response data are presented in Figure 26 as $\Delta P_{vd}/\Delta P_{cd}$ (valve differential pressure/input differential pressure) versus frequency and $\Delta V/\Delta P_{vd}$ (ram velocity/valve differential pressure) versus frequency and $\Delta V/\Delta P_{cd}$ the total system response. These three responses were measured independently and the summation of the amplifier and valve responses to equal the total response is a check on the experimental method. The amplitude response data show that the fluid amplifier increasing gain with frequency characteristic in the 0 to 130 Hz range combines with the decreasing gain of the valve to provide an overall amplitude that has decreased to 70% of its static gain at 30 Hz with the major reduction in gain due to the first and second stages of the valve. The phase characteristics show that at 30 Hz the

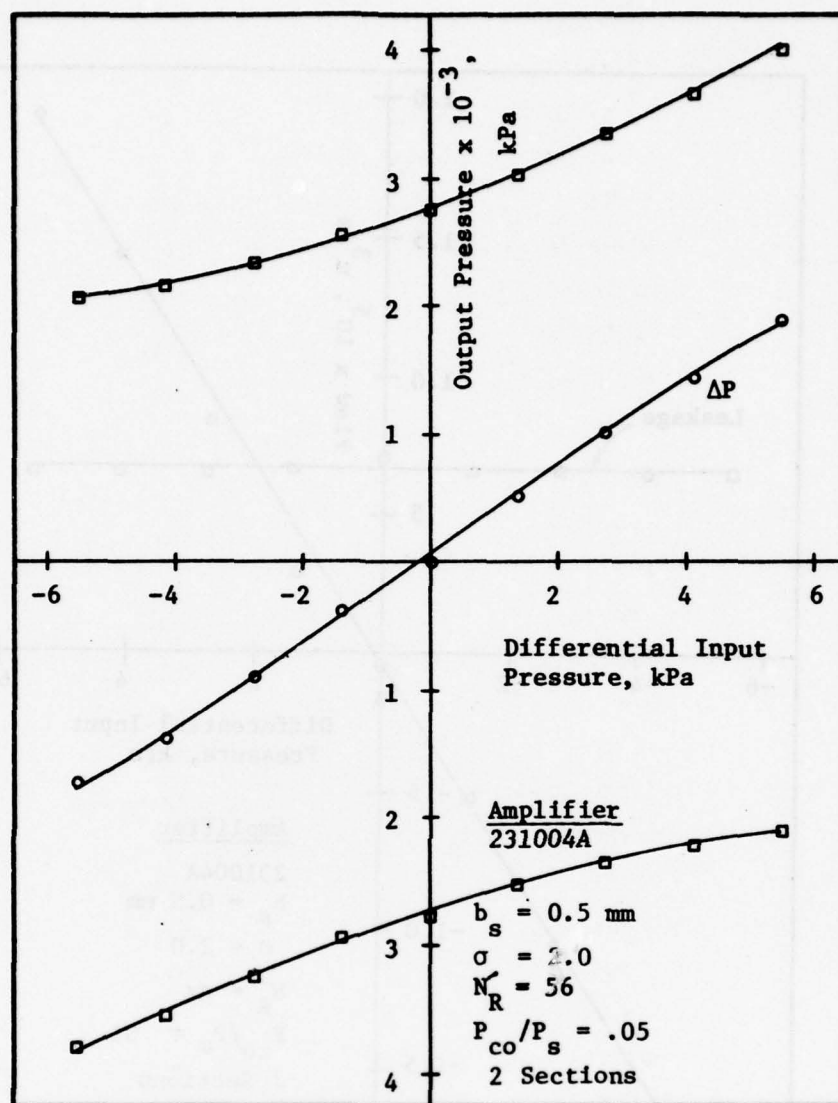


FIGURE 24. FLUIDIC INPUT VALVE PRESSURE GAIN

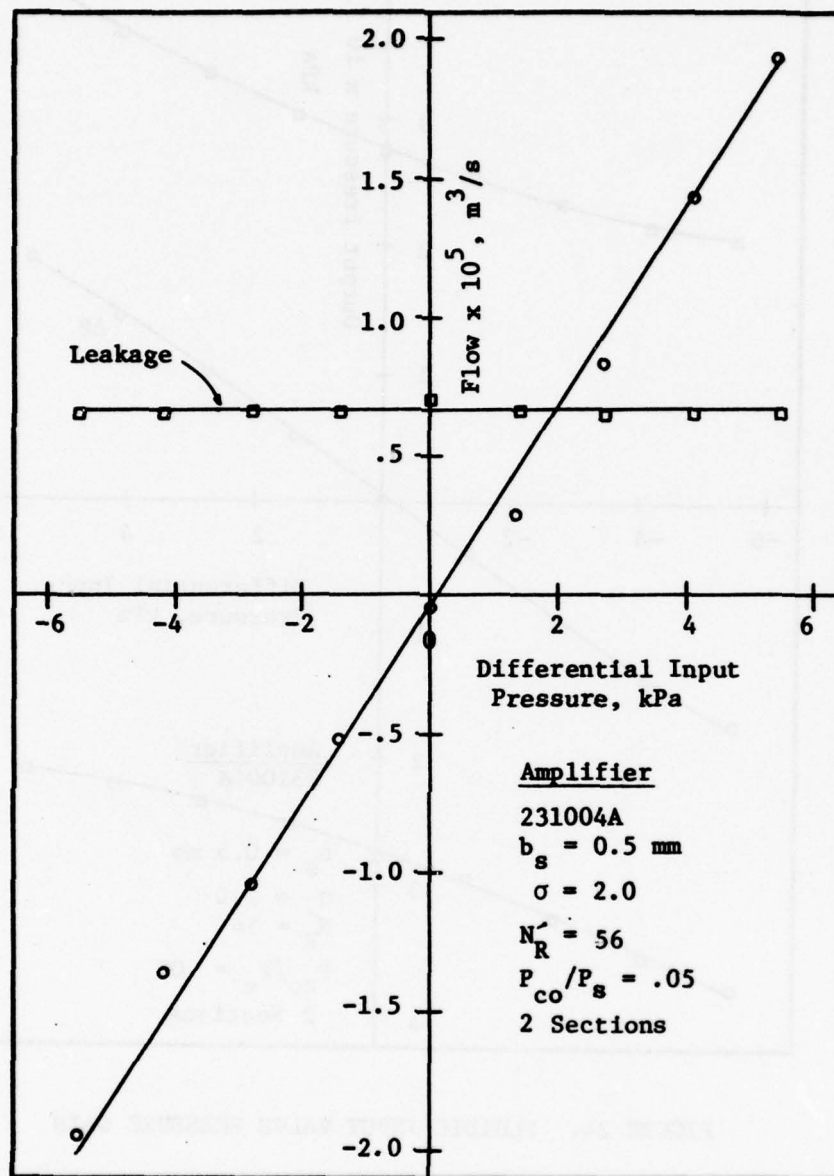


FIGURE 25. FLUIDIC INPUT VALVE FLOW VERSUS AMPLIFIER INPUT

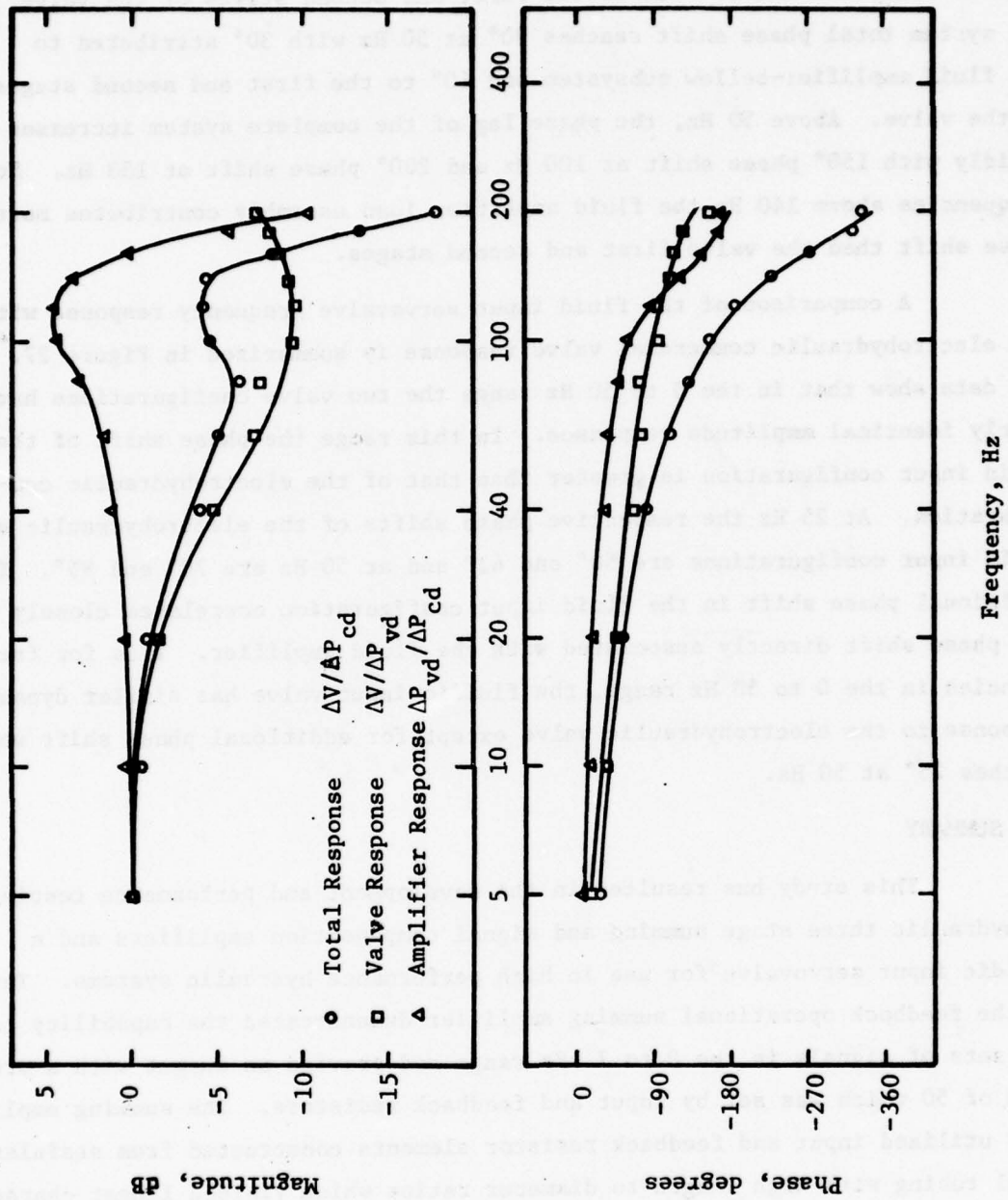


FIGURE 26. FLUIDIC INPUT VALVE FREQUENCY RESPONSE

total phase shift is 70° with 20° degrees attributed to the fluid amplifier-bellows subsystem and 50° due to the first and second stages of the valve. The system total phase shift reaches 90° at 50 Hz with 30° attributed to the fluid amplifier-bellow subsystem and 60° to the first and second stages of the valve. Above 50 Hz, the phase lag of the complete system increases rapidly with 150° phase shift at 100 Hz and 200° phase shift at 150 Hz. For frequencies above 140 Hz the fluid amplifier load assembly contributes more phase shift than the valve first and second stages.

A comparison of the fluid input servovalve frequency response with the electrohydraulic commercial valve response is summarized in Figure 27.* The data show that in the 0 to 50 Hz range the two valve configurations have nearly identical amplitude responses. In this range the phase shift of the fluid input configuration is greater than that of the electrohydraulic configuration. At 25 Hz the respective phase shifts of the electrohydraulic and fluid input configurations are 50° and 62° and at 50 Hz are 70° and 95° . The additional phase shift in the fluid input configuration correlates closely with the phase shift directly associated with the fluid amplifier. Thus for frequencies in the 0 to 50 Hz range, the fluidic input valve has similar dynamic response to the electrohydraulic valve except for additional phase shift which reaches 25° at 50 Hz.

4. SUMMARY

This study has resulted in the development and performance testing of hydraulic three stage summing and signal compensation amplifiers and a fluidic input servovalve for use in high performance hydraulic systems. Tests of the feedback operational summing amplifier demonstrated the capability to sum sets of signals in the 0 to 7 kPa range and provide an output with a pressure gain of 50 which was set by input and feedback resistors. The summing amplifier utilized input and feedback resistor elements constructed from stainless steel tubing with high length to diameter ratios which yielded linear characteristics over the operating range of interest. Tests conducted on the amplifier demonstrated both input-output linearity and superposition of inputs.

* The electrohydraulic valve response with the torque motor input was measured using a piston for flow measurement in a similar manner to that described above. The response was measured by Moog Valve Company.

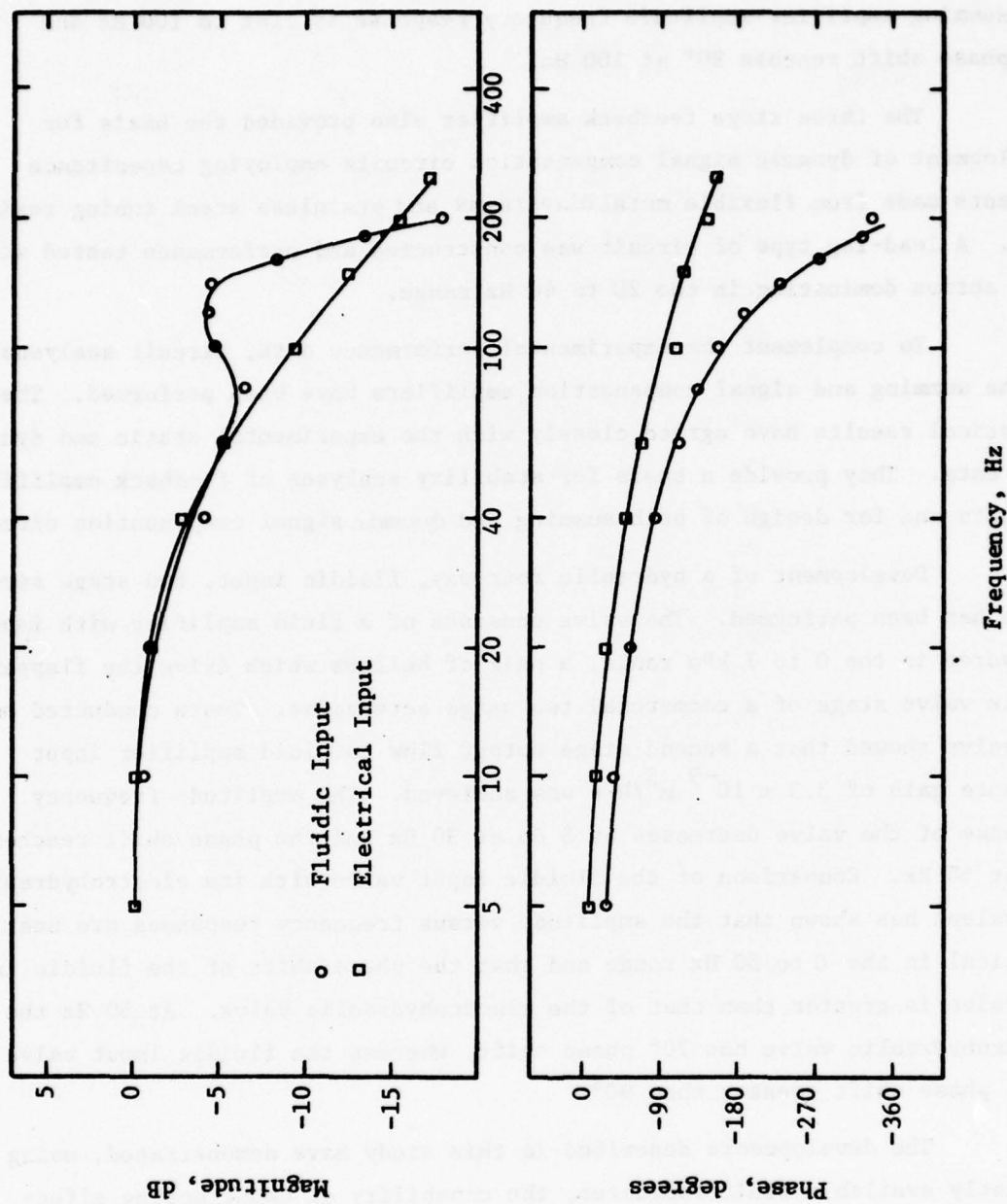


FIGURE 27. COMPARISON OF FLUIDIC AND ELECTROHYDRAULIC VALVE RESPONSES

The summing amplifier amplitude frequency response is flat to 100 Hz and the phase shift reaches 90° at 100 Hz.

The three stage feedback amplifier also provided the basis for development of dynamic signal compensation circuits employing capacitance elements made from flexible metal diaphragms and stainless steel tubing resistors. A lead-lag type of circuit was constructed and performance tested with lead action dominating in the 20 to 40 Hz range.

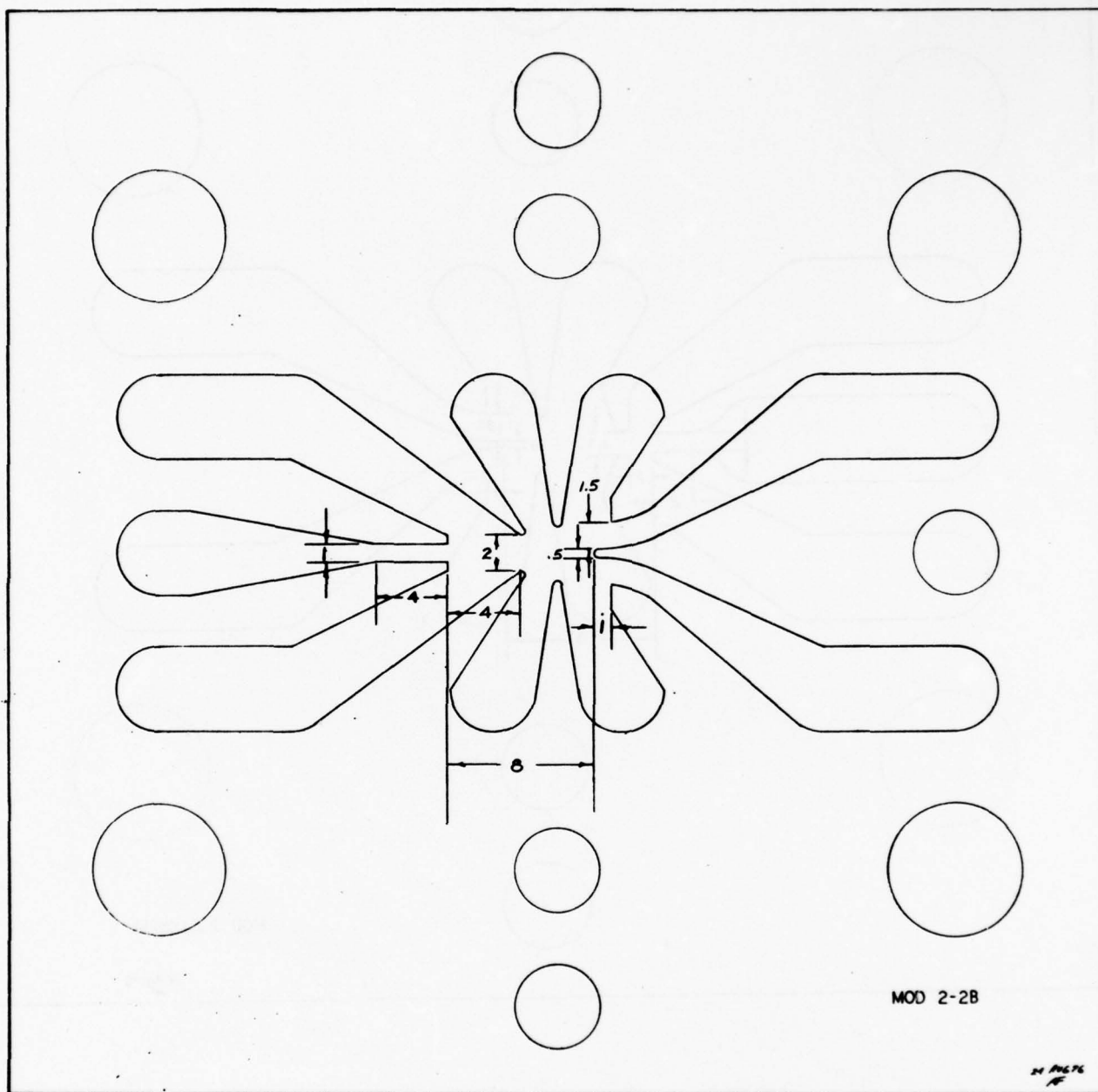
To complement the experimental performance data, circuit analyses of the summing and signal compensation amplifiers have been performed. These analytical results have agreed closely with the experimental static and dynamic test data. They provide a basis for stability analyses of feedback amplifier circuits and for design of both summing and dynamic signal compensation circuits.

Development of a hydraulic four-way, fluidic input, two stage servovalve has been performed. The valve consists of a fluid amplifier with input pressures in the 0 to 7 kPa range, a pair of bellows which drive the flapper-nozzle valve stage of a commercial two stage servovalve. Tests conducted on the valve showed that a second stage output flow to fluid amplifier input pressure gain of $3.3 \times 10^{-9} \text{ m}^5/\text{N-s}$ was achieved. The amplitude frequency response of the valve decreases by 3 db at 30 Hz and the phase shift reaches 90° at 50 Hz. Comparison of the fluidic input valve with its electrohydraulic equivalent has shown that the amplitude versus frequency responses are nearly identical in the 0 to 50 Hz range and that the phase shift of the fluidic input valve is greater than that of the electrohydraulic valve. At 50 Hz the electrohydraulic valve has 70° phase shift, whereas the fluidic input valve has a phase shift greater than 90°.

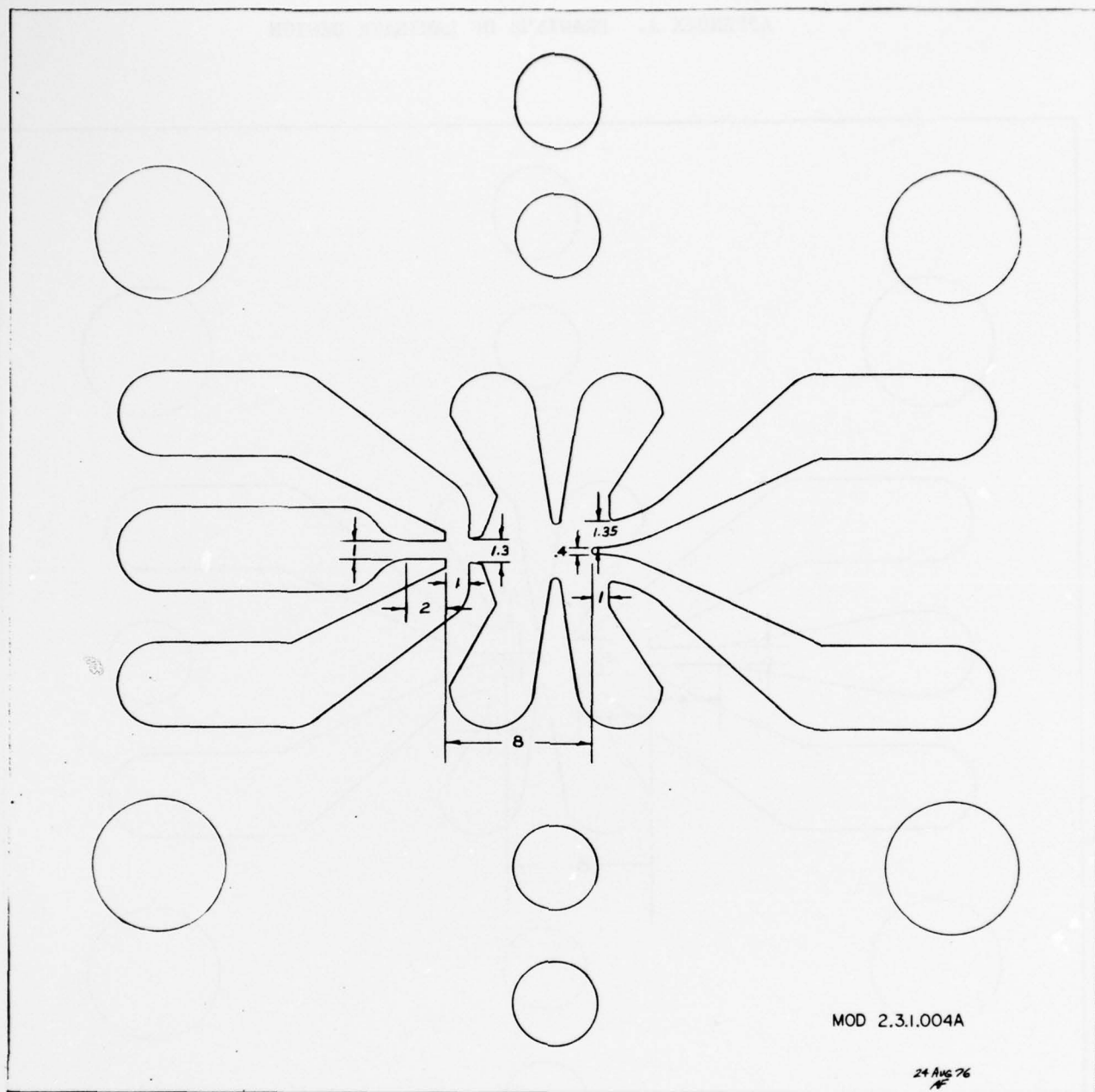
The developments described in this study have demonstrated, using currently available fluid laminates, the capability of constructing effective summing and signal compensation elements in a hydraulic medium. These elements, with the fluid input servovalve provide a capability to construct closed-loop, high power level hydraulic systems in which sensing, signal compensation and power level modulation functions are performed with hydro-mechanical components.

The research described in this report completes the second phase of work directed to high performance hydraulic system development. Work in the next phases of research will focus upon integration of components into a complete system and upon the problems associated with operation of hydraulic systems over wide ranges in temperature.

APPENDIX A. DRAWINGS OF LAMINATE DESIGN



2-2B LAMINATE SCALED IN NOZZLEWIDTHS



231004A LAMINATE SCALED IN NOZZLEWIDTHS

NOMENCLATURE

A_i	tube area of resistor i (mm^2)
b_s	supply nozzle width (mm)
C	capacitance (m^5/N)
d_i	tube diameter of resistor i (mm)
h	laminate thickness (mm)
I_i	inertance of resistor i ($\text{N-s}^2/\text{m}^5$)
K	static gain
l_i	tube length of resistor i (mm)
N_R	Reynolds number
N'_R	modified Reynolds number
P_{bd}	bellows pressure differential (kPa)
P_c	control port pressure (kPa)
P_{cd}	control port pressure differential (kPa)
P_{co}	control port pressure bias ($(P_{c1l} + P_{c1r})/2$) kPa
P_{fd}	feedback pressure differential (kPa)
P_i	input pressure (kPa)
P_o	output port pressure, output pressure (kPa)
P_{od}	output port pressure differential (kPa)
P_{vd}	valve input pressure differential (kPa)
P_s	supply pressure (kPa)
P_1, P_2	summing amplifier input pressure (kPa)
P_{1d}, P_{2d}	summing amplifier input pressure differential (kPa)
Q_c	control port flow (m^3/s)
Q_o	output port flow (m^3/s)
Q_s	power jet supply flow (m^3/s)
R	resistance ($\text{N-s}/\text{m}^5$)
R_D, R_{D1}, R_{D2}	pressure divider resistance ($\text{N-s}/\text{m}^5$)
R_f	feedback resistance ($\text{N-s}/\text{m}^5$)
R_g	grounding resistance ($\text{N-s}/\text{m}^5$)
R_i	resistance of resistor i ($\text{N-s}/\text{m}^5$)
R_o	amplifier output resistance ($\text{N-s}/\text{m}^5$)
R_1, R_2	resistors ($\text{N-s}/\text{m}^5$)

R^*, R^{**}	resistance as defined in eq. (4)
s	Laplace transform operator
T	output-input pressure transfer function
T_1	first order lag time constant (s)
V	ram velocity (m/s)
x_{th}	supply nozzle throat length (mm)
Z_a	amplifier input impedance ($N\text{-s/m}^5$)
Z_{D1}, Z_{D2}	pressure divider impedance ($N\text{-s/m}^5$)
Z_f	feedback resistor impedance ($N\text{-s/m}^5$)
Z_g	grounding resistor impedance ($N\text{-s/m}^5$)
Z_i	impedance of resistor i ($N\text{-s/m}^5$)
Z_o	output impedance ($N\text{-s/m}^5$)
Z_1, Z_2	input resistor impedance ($N\text{-s/m}^5$)
$\Delta()$	increment or difference of a quantity ()
μ	absolute viscosity
ξ	second order factor damping ratio
ρ	fluid density (kg/m^3)
σ	amplifier power nozzle aspect ratio $[h/b_s]$
τ	delay time constant (s)
τ_i	time constant of resistor i (s)
τ_1, τ_2	time constants (s)
ω_n	second order factor natural frequency (rad/s)

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